THE ANTI-ROLL STABILIZATION OF SHIPS

Jose Roberto Escalona

United States
Naval Postgraduate School

THESIS

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by

Jose Roberto Escalona

Thesis Advisor: G. J. Thaler

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Jose Roberto Escalona
Lieutenant Commander, Venezuelan Navy
B.S.E.E., Naval Postgraduate School, 1970

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ABSTRACT

The theory of Roll stabilization of Ships is presented in the context of modern control theory. The most common systems used to reduce the roll are described, and the principal equations are formulated.

A general approach for the analysis of roll stabilizers is developed and it is applied to an activated fin stabilizer system. For this approach parameter plane techniques were applied, and the system was simulated in the Digital Computer by means of the Continuous System Modeling Program CSMP-IBM/360.

Finally a system is proposed which is intended to improve the performance of passive tank stabilizers introducing fluidic devices in the feedback loop in addition to a supply of air compressed to actuate on the water ballast. The system was simulated using the same program CSMP-IBM/360, and the results compared with those obtained in the simulation of a simple passive tank stabilizer, showing a significant increasing in the damping.

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I. INTRODUCTION

A. DISCUSSION

The problem of stabilizing ships against the different motions in a seaway has been studied for many years and is not at the present completely solved even though great advances have been made in the military and commercial construction of ships.

In commercial shipbuilding, stabilization of motions is important especially in passenger ships and the main reason is to provide passenger comfort. Aboard military ships it is also important because it can improve performance of search radar, sonar, weapon systems, crew's efficiency and some other routine tasks. There exists a considerable number of works concerning the motion of ships and numerous means have been developed to reduce the undesired motion. Since 1875 this subject has been under consideration and it was at that time when W. Froude [1] presented a paper to the Institution of Naval Architects. In that paper is described a method of graphical integration suited to compute the resultant motion of a ship, starting from a presumed train of waves and a given vessel's stability characteristics. The theory developed by W. Froude has served as a foundation for further studies and investigations about Rolling motion and the appropriate methods to reduce it. Other authors also have contributed in this field and their contributions are listed in the Bibliography.

Theoretically a ship floating on the surface of quiet or disturbed water may be considered as a rigid body performing oscillations that constitute the motions of the ship.

The motions of a ship have been observed in three fundamental forms, which are: rolling, pitching and heaving motions.

1. Rolling

It is distinguished as a rotational oscillatory motion of the ship about a longitudinal axis.

2. Pitching

Consists in a rotational oscillatory motion of the ship about a transverse axis.

3. Heaving

Consists in a translatory oscillatory motion of the ship in the vertical direction.

The last two motions, heaving and pitching, have been treated in the past without any success and the results have lead to the conclusion that in order to reduce these motions, it is necessary to introduce drastic changes in the conventional forms of ships.

Rolling motion, however, can be largely reduced with very small changes in the structure of ships. A change in the structure has been the installation of bilge keels which are fitted in the submerged part of the hull of the ship.

Rolling was defined as a rotational oscillatory motion about a longitudinal axis, therefore, the position of the ship is given by its angle of inclination ϕ.

Figure 1 illustrates the three fundamental ship motions defined above.

Considerable effort has been devoted to the study of rolling of ships and the main objective of this thesis is to provide the reader with a comprehensive theory about rolling on ships and its effects, describes the most important systems to reduce the rolling actually in use and their characteristics, realize a digital computer simulation and finally write the necessary conclusions and recommendations in order to establish a suitable system for practical purposes.

B. MOTION STABILIZERS

Stabilizers are generally distinguished in accordance with two fundamentals:

1. According to the origin of the forces that produce the restoring moment; in this category are included the following groups:
    a. Stabilizers whose stabilizing moment is created by means of weight forces.
    b. Stabilizers whose stabilizing moment is created by the action of hydrodynamic forces.
    c. Stabilizers whose stabilizing moment is created by the action of gyroscopic forces.

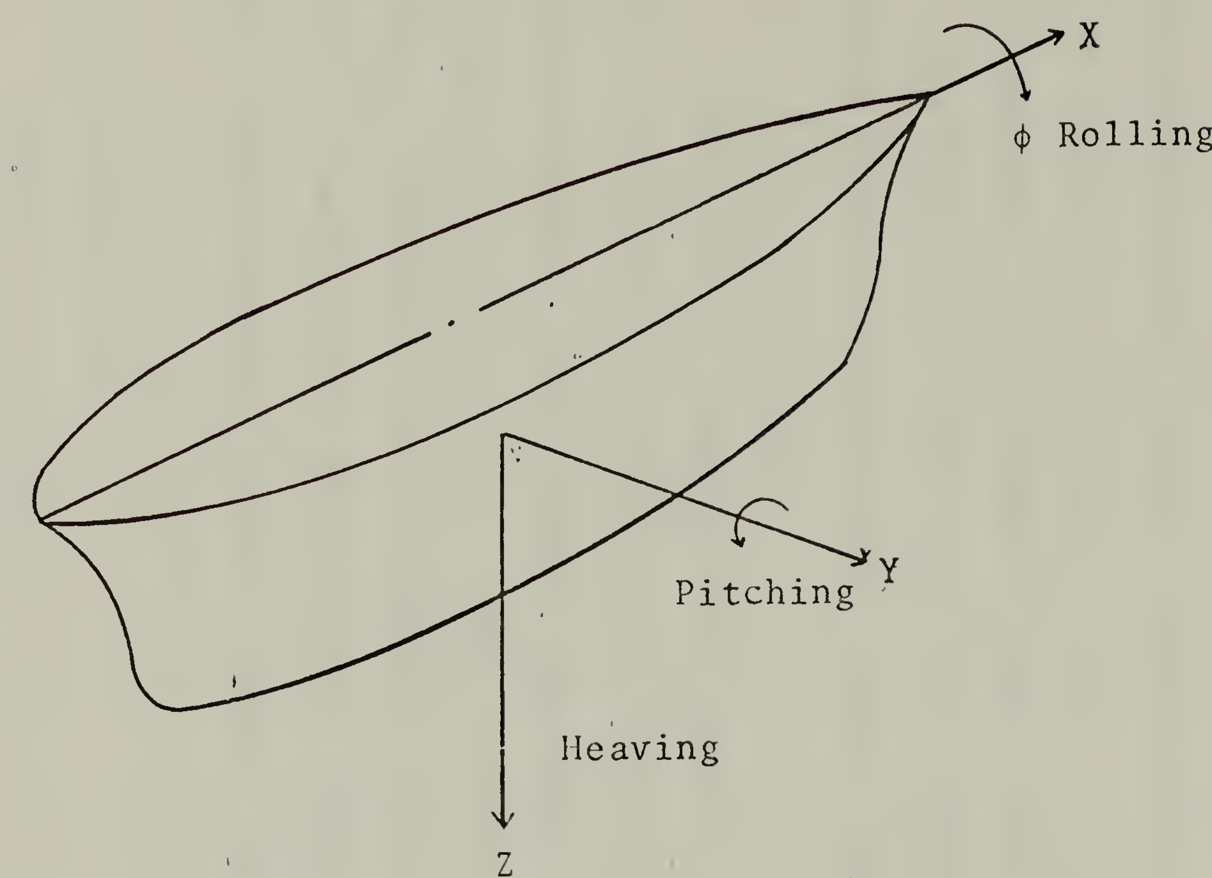

Figure 1. Ship Motions and Coordinates.

2. <u>According to the principle of operation, this category includes the following groups</u>:

a. Passive Stabilizers

These types of stabilizers do not require additional power for their operation, they are constrained only by space and weight, are not controlled by any automatic device actuated by disturbance signals and could be considered as special cases under the more general class of active stabilizers, in whose case they correspond to active systems with zero gain in the control loops.

b. Active Stabilizers

This class of stabilizers requires power and control units in order to operate the appropriate devices designed to produce an added stabilizing moment Ms, the effect of which is to result in a reduction of the rolling motion. Active stabilizers of course are more expensive and require more maintenance than passive stabilizers but they are also more effective in reducing the roll. In general, in any event, every active device has its corresponding passive counterpart.

The next section gives a brief explanation of some of these types of stabilizers and in further chapters will be detailed the operation and characteristics of the most useful systems.

C. DESCRIPTION OF THE MOST USEFUL STABILIZERS

After the work presented by Froude, rolling was intended to be reduced primarily by using the simplest type

of stabilizers, which are known as "Bilge Keels," these consist of long plates which are fitted along the line of flow near the bilge of the ships.

Bilge keels are widely applied in many ships and are normally fitted to vessels because they have a significant stabilizing effect. These keels must have sufficient strength in order to resist the pressure of the water during motions. Among the advantages of using bilge keels can be mentioned, their low cost, their very simple construction and the fact that they do not occupy any useful space within the ship, however, their operation is most effective only when they are operating in a regime close to resonance.

Figure 2 illustrates a bilge keel on a ship.

When it is decided to install bilge keels in a ship, the problem is generally solved on the basis of empirical calculations or on the basis of models tested in a tank. In general, the dimensions are given by the following specifications: the total area of the keel is taken from 2 to 4 percent of the product of the vessel length L and the breadth B; the height is from 0.3 to 1.2m, depending upon the type of ship, on the average is from 3 to 5 percent of the vessel beam and the length is from 25 to 75 percent of the length of the ship.

Besides the use of bilge keels as stabilizers, during recent years three types of roll stabilizers have been used and have demonstrated a significant amount of success, they

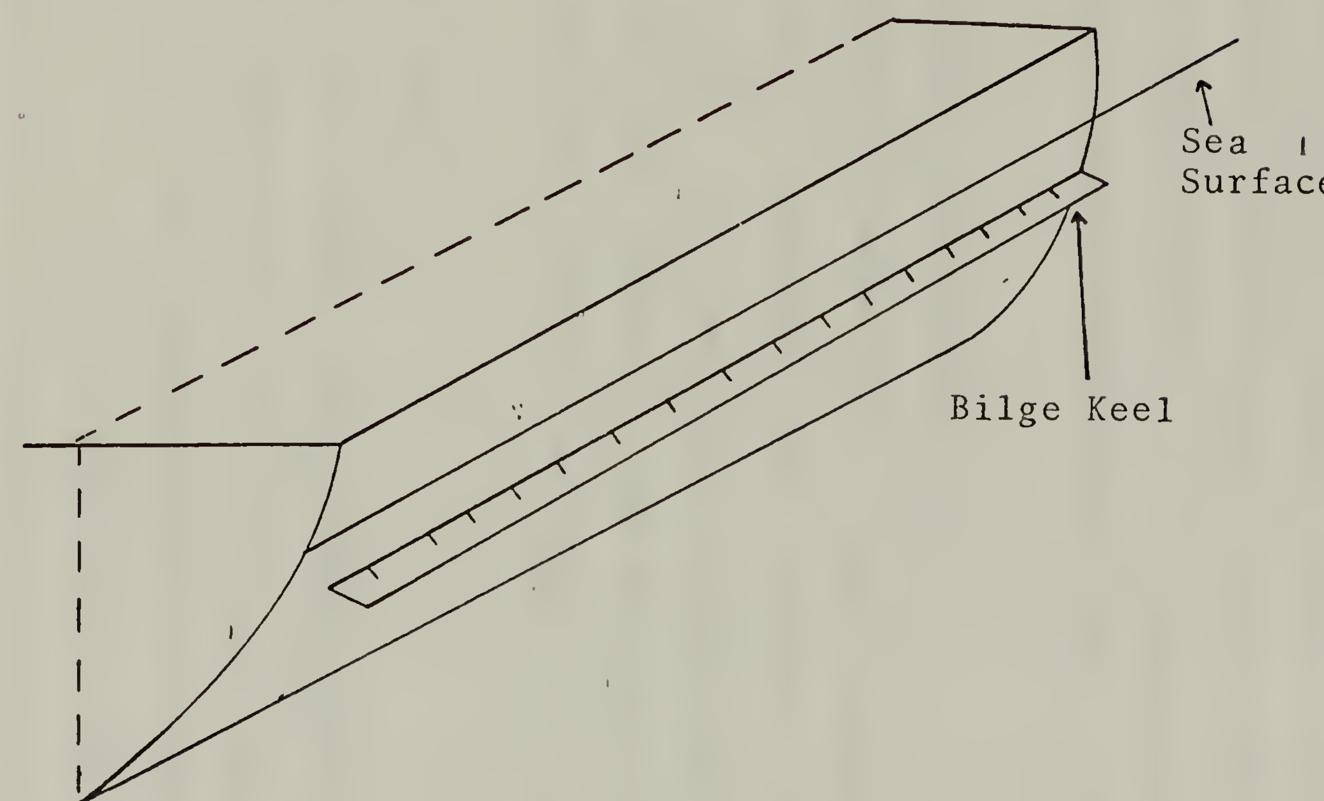

Figure 2. Bilge Keel Configuration.

are tank stabilizers, activated fins stabilizers, and gyro-stabilizers.

1. Tank Stabilizers

These are fluid reservoirs built and suitably located on board of ships in such a way that when the ship is affected by rolling motion, the fluid moving inside the tank, produces a moment whose action tends to stabilize the ship.

2. Activated Fins Stabilizers

These are controlled underwater wings projecting from both sides of the ship, near the center, and may be rotated on stems. The stems pass through water-tight stuffing boxes into the interior of the hull and their rotation is done by means of a special, automatically controlled drive.

3. Gyrostabilizers

These are compact units located generally below decks on the central axis of the ship and consist basically of a larger rotor with a single degree of freedom about an arthwartship axis. Besides the spinning movement of its rotor the gyro exerts a righting force against the rolling motion tending in this manner to stabilize the ship.

D. CONCLUSIONS

In designing a ship, several factors influence the decision to be taken for the installation of passive or active stabilizers. Cost is one of those factors and it is

desired to pay a minimum price for devices that guarantee to keep the motions within acceptable limits. For instance, a ship equipped originally with a stabilizing system having sufficient size to overcome the most severe sea conditions could operate in a very uneconomic mode under less severe sea states, therefore it is necessary to have some idea of the range of sea states that the ship might encounter.

In order to specify a suitable stabilizer for any particular ship, modern concepts of ship motion, sea state and control theory will be discussed in the following chapter.

II. LINEAR THEORY OF ROLLING

A. DISCUSSION

The superposition principle can be applied to any sea surface. Simple waves, each having a particular wavelength, direction of advance and having no interference with one another, form a complex wave surface and a ship on that surface responds independently to the action of each individual wave, its resultant motion being the sum of all the responses to each individual wave. The magnitude of the ship response is proportional to the height of the waves, however there exists some objection to the application of this theory to rolling motion, that is that the response of models to waves of a given length and direction has shown that they are not proportional to the height of the wave. This problem has lead to nonlinear theories [2] which have not been successful because of their cumbersome solution. For this reason linear theory has been developed which might give a reasonable engineering representation of the problem of rolling motion.

In the study of rolling motion it is necessary to assume that the plane of symmetry of the ship is parallel to the crests of the approaching waves. Another assumption is that the center of gravity of the ship moves in a circular orbit in a vertical plane and the diameter is equal to the height of the wave.

Other considerations in this study are the different forces acting upon a ship which is rolling in a seaway: a diagram representing these forces is shown in Figure 3 as given by N. Blagoveshchensky [3].

B. EQUATION OF ROLLING

Refer to Figure 4 for the formulation of the general equation for unstabilized rolling motion of a ship, as given in Reference 2. A right-hand system of coordinate axes will be adopted and designated as Gxyz being fixed in the ship, the origin is the center of gravity G of the ship, the axes are Gx, Gy and Gz along the principal axes of inertia of the ship's mass. The positive direction of the axes are taken as follows: Gx horizontally pointing toward the bow, Gy horizontal to starboard and Gz vertically directed downwards.

It is assumed that the transverse dimensions of the ship are small in comparison with the length and height of the wave.

This assumption permits us to consider the vessel as a particle of the wave.

Under the given assumptions consider a small element of the ship as shown in Figure 4, the restoring force from this element is acting in the direction of the apparent vertical which is perpendicular to the wave surface and the force is given by

$$f_r = \rho g \, ds \, dx \qquad \text{(II.1)}$$

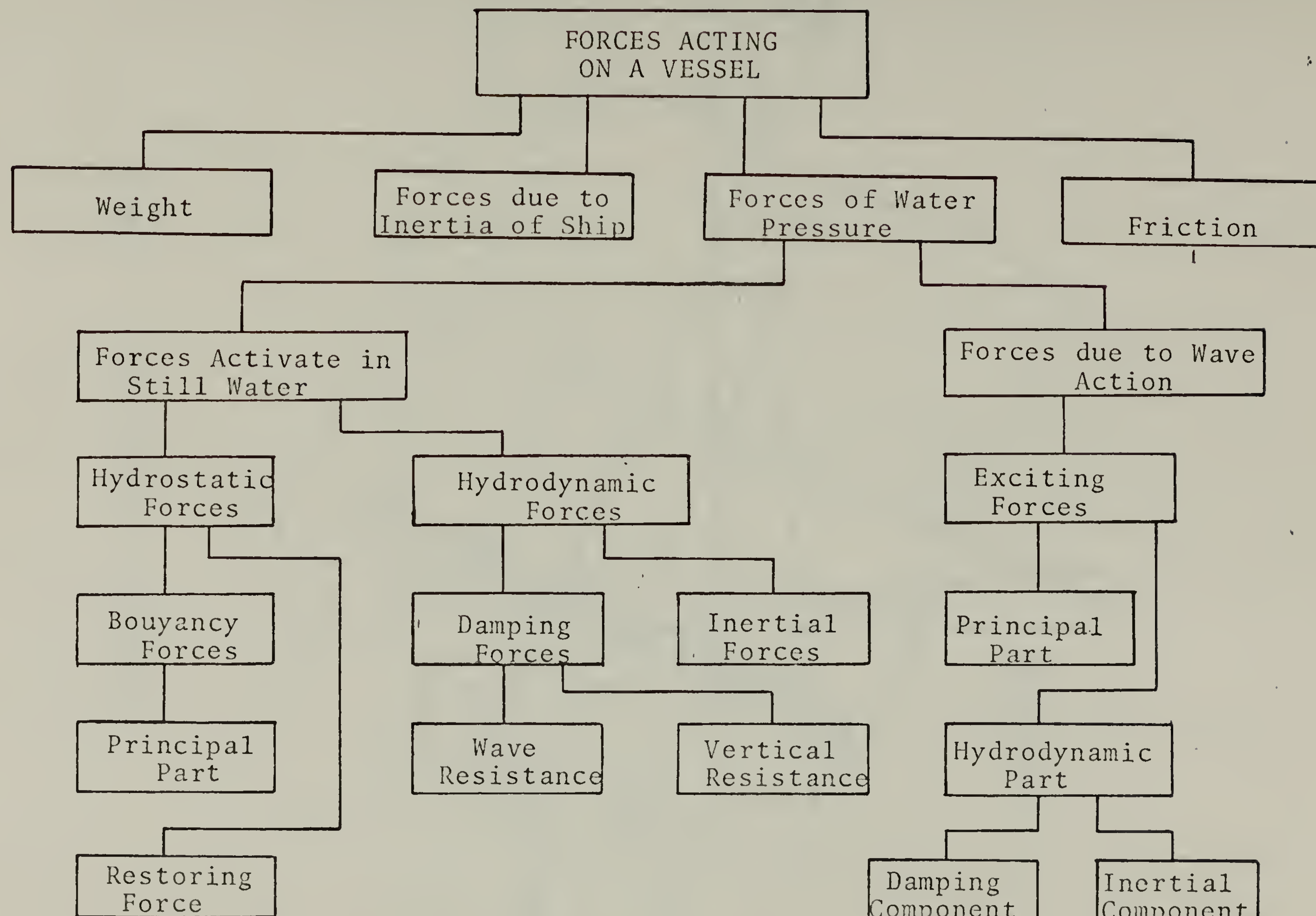

Figure 3. Forces Acting on a Vessel.

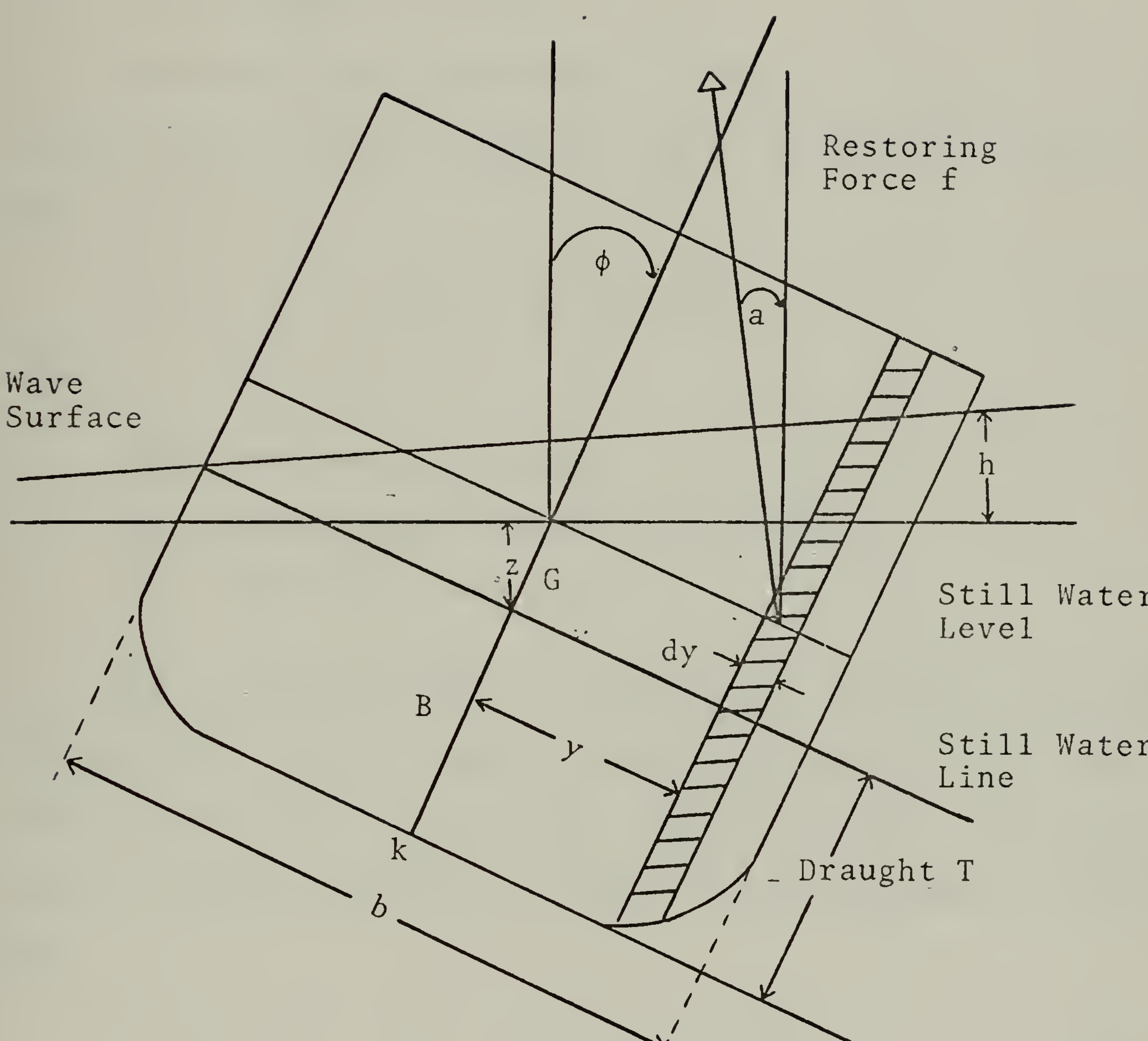

Figure 4. Cross Section of Ship.

where

ρ = Density of water

g = Acceleration of gravity

ds = Area of strip

dx = Length of strip.

The total restoring moment on this slice about a horizontal axis through the ship's center of gravity is given by:

$$Mr = -\overline{GM}\ W(\phi + a_1) \qquad \text{(II.2)}$$

where

$\overline{GM}$ = Metracentric height

W = Displacement weight of the ship

a_1 = Effective wave slope.

The structural moment of inertia of the ship increases effectively when the ship is rolling relative to the sea, this is due to the acceleration imparted to the surrounding water, and the expression for this inertial moment is given by:

$$Mi = I\ddot{\phi} + \delta I\ (\ddot{\phi} + a_2) \qquad \text{(II.3)}$$

where

I = Rolling moment of inertia of the ship

δI = Moment of inertia of the entrained water

a_2 = Effective wave slope (not necessarily the same as a_1).

The relative angular velocity between ship and water originates a damping moment that is given by:

$$Md = -2N\,(\dot{\phi} + a_2) \qquad \text{(II.4)}$$

where

N = Damping coefficient

2N is the damping moment when the angular velocity relative to the surrounding water is equal to unity.

ϕ, $\dot{\phi}$, and $\ddot{\phi}$ are roll angle, roll velocity and roll acceleration respectively.

Combining Equations (II.2), (II.3), and (II.4), a general equation for unstabilized rolling motion in a ship is obtained which is given by:

$$(I + \delta I)\,\ddot{\phi} + 2N\dot{\phi} + \overline{GM}\,W\,\phi + \delta I\ddot{a}_2 + Na_2 + \overline{GM}\,Wa_1 = 0 \qquad \text{(II.5)}$$

Equation (II.5) considers the heading of the ship relative to the advance of the waves, whose primary effect is to vary the slope and effective frequency of the waves meeting the ship.

C. THE CONTROL ACTION IN ROLL STABILIZATION

It is necessary to introduce a control system capable of automatically driving the stabilizer when it is required to overcome the rolling action.

The control systems for stabilizers may be classified as "continuous" and "discontinuous" [4]. This classification depends upon the conditions of the control signal relative to the regulated quantity, that is, in continuous control

the regulated quantity follows proportionally the control signal while in discontinuous control the regulated quantity takes extreme negative or positive values depending on the sign of the control signal. In general when using continuous control, the amount of power required to position the stabilizer will be lower than when using discontinuous control.

Basically there exist two modes of control in a stabilizer system, they are "feedback" control and "feedforward" control and those denominations are given according with the signal that derives the control action. In feedback control the signal is taken from the vertical inclination of the ship while in feedforward control the signal is taken in some manner from sea slope or some other input.

The majority of installations are equipped with a feedback control system, the reason for this is that it is easier to sense ship motion than to sense wave slope. The only advantage of using feedforward control over feedback control is that the problem of self-oscillations due to closed loop is avoided, but in spite to this problem feedback systems are relatively independent of variations in some system parameters e.g. ship displacement, metacentric height, velocity, etc.

Figure 5 shows a block diagram of a stabilizer feedback control system including the ship, control unit and sensors, and the stabilizers.

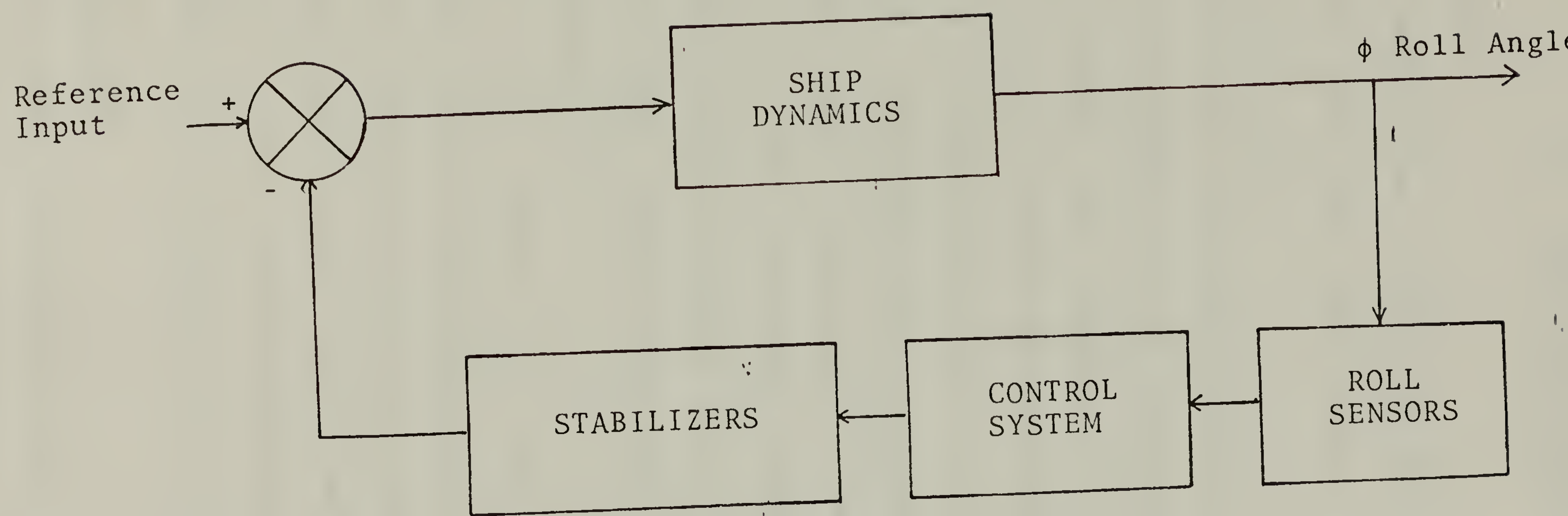

Figure 5. Block Diagram Feedback Control Stabilizer System.

III. TANK STABILIZERS

A. GENERAL

The first system of anti-rolling tank was installed on a ship about 1874 [12], a few years after the first installation of bilge keels.

The earliest applications consisted in reservoirs installed in the upper part of the vessel where free water could be carried. This configuration created a reduction in the metacentric height of the vessel due to the added weight high and also due to the free surface of the water in the tank, the result was, an increasing in the ship's roll period. Also if the ship has been rolling in synchronism with the waves, the equality of periods disappears, and the displacement of water to the low side of the ship produced a moment opposing the ship's righting moment thereby damping the roll.

This "water chambers," as they were called, are no longer used because they bring some problems. They are potentially dangerous because any reduction in rolling is obtained by reduction of righting moment, and when the water in the tanks, and the vessel get into synchronism the water could help to increase the rolling action.

B. DISCUSSION

There exist two types of tank stabilizers:

1. Passive Tank Stabilizers.
2. Active Tank Stabilizers.

The passive type can be defined as fluid pendulum-dynamical absorbers of oscillations. The displacement of considerable amount of liquid from one side compartment to another produces a stabilizing moment, which is generated by the action of weight forces.

The active type operates under the same principle as the passive type but the difference is that in this system the fluid is moved by means of some activating agents such as blowers and propeller pumps.

Any type of tank stabilizer has its own inherent characteristics. For example it can be observed that a tank filling with fluid needs some time to accumulate its complete effectiveness. But on the other hand there are some type of stabilizers that generate the required force faster because that force is produced directly and they can be adjusted rapidly due to their smaller mass, as an example consider the activated fin stabilizer which will be described later.

Different types of tank stabilizers are available and the principle of operation of all of them is analogous. Figure 6 shows the section of a tank stabilizer system.

Other types of tank stabilizers that have been used in the past are illustrated in Figure 7.

C. PASSIVE TANK STABILIZERS

The passive tank stabilizer is characterized by the fact that the fluid is forced from one side of the ship to the other by the ship's motion. It has been claimed that

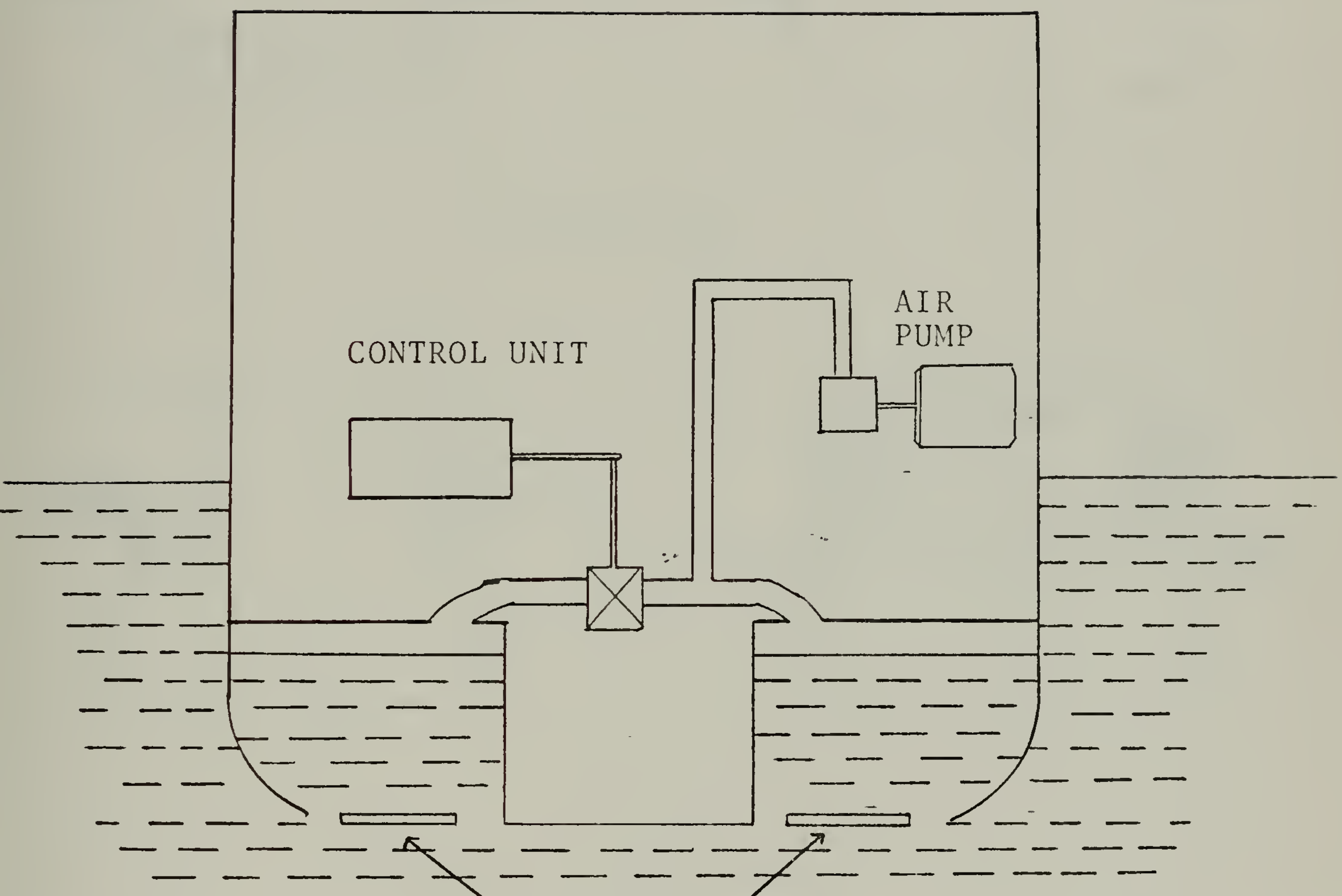

Figure 6. Section of a Tank Stabilizer.

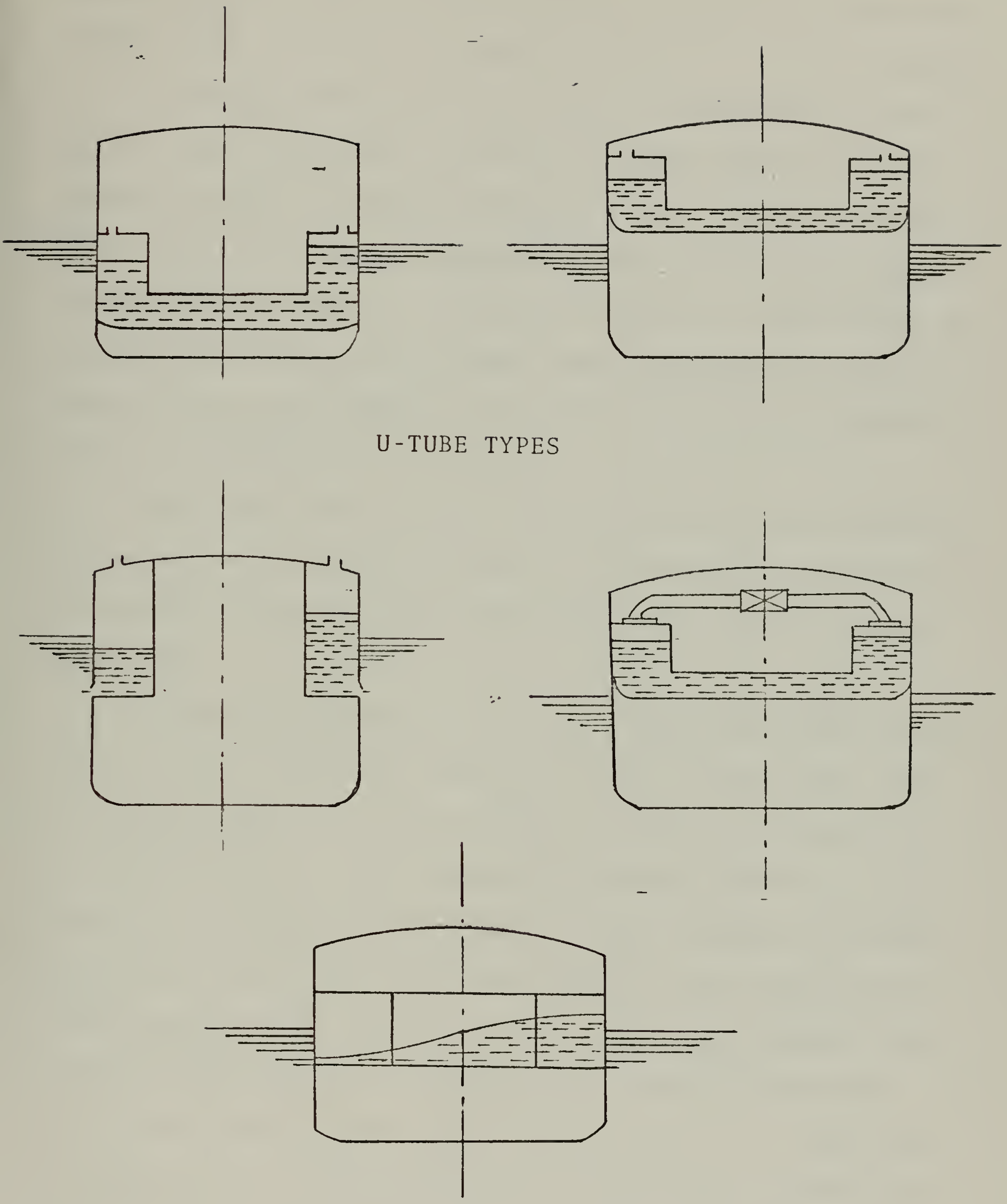

Figure 7. Typical Tank Stabilizer Configurations.

using a passive tank stabilizer the rolling can be reduced in the order of 60 to 70 percent.

As already mentioned the theory of operation of the passive tank stabilizer is based upon the theory of the undamped dynamic vibration absorber; this theory says that if a secondary resonant system is connected to a vibrating or oscillating body at the natural period of oscillation of the primary system, it is found that the motion of the secondary system lags behind that of the main body by approximately a quarter of a cycle [6].

This theory will be best understood observing the different sequences shown in Figure 8 representing the motion of liquid in passive tank stabilizer under steady rolling condition.

It can be observed in the figure that the fluid inside the port and starboard tanks oscillates in such a way that the ship's period of roll is tuned so that the motion of the liquid tends to lag a quarter of cycle behind the roll. That is, a maximum displacement of fluid between the tanks occurs when the ship is at the greatest angle from the vertical and the flow reverses as the ship passes through the vertical when the angular velocity of roll is a maximum.

In designing passive tank stabilizers it is important to ensure that a convenient phase relationship between the motion of the liquid and the roll should be sustained over a sufficiently wide range of rolling periods. According to that, in some cases it is convenient to introduce appropriate

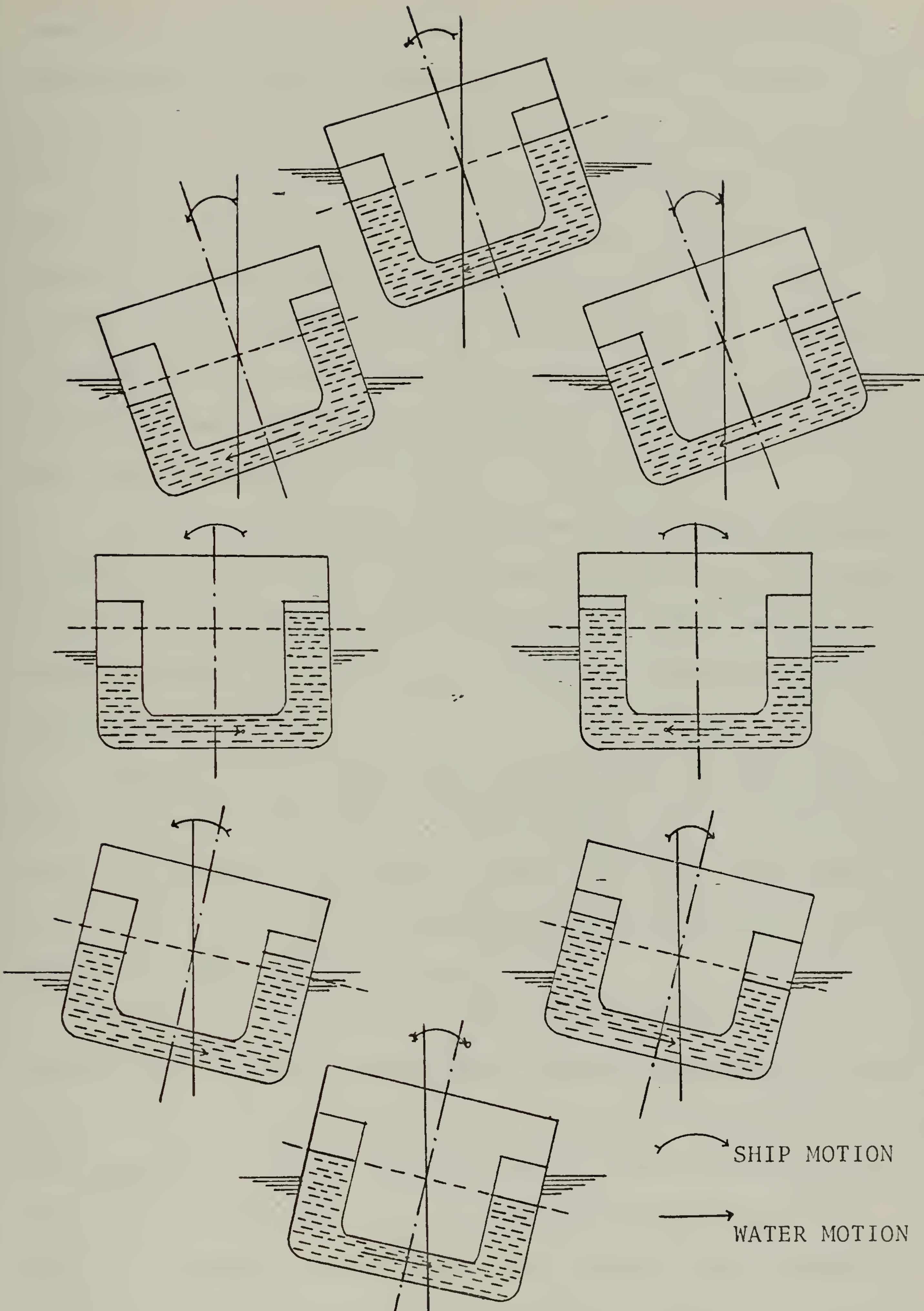

Figure 8. Combined Motion of Ship and Water Ballast.

means for adjusting the natural period of oscillation of the liquid to adapt the period of roll under different sea conditions. A practical means to avoid large amplitude rolling at long periods is to obstruct in part the flow of fluid between the tanks, this can be done using some configuration that could provide eddies in the flow; this method reduces considerably the efficiency of the stabilizer in some cases from 90 percent to 60 percent and, in consequence at long periods the tank reduces its ability to roll the ship (in error).

Figure 9 shows some curves taken from a paper written by Wollard [7] in 1913, in the curves is observed a reverse rolling due to the tank system both above and below the natural period of the ship, but excellent stabilization at the natural period.

Figure 10 also represents a typical set of free roll and stabilized curves for a ship provided with a system of uncontrolled tank stabilizer, these curves give the practical results obtained by using an analog table which is described in detail in Reference 5.

As far as is known, the theory for prediction of the performance of free surface tanks without experimental data has not been adequately developed. Much of the theory has been concerned with the calculations of the natural period for tank configurations of varying degrees of complexity, but the internal damping of the oscillating fluid has been given less importance. Theories have been developed which

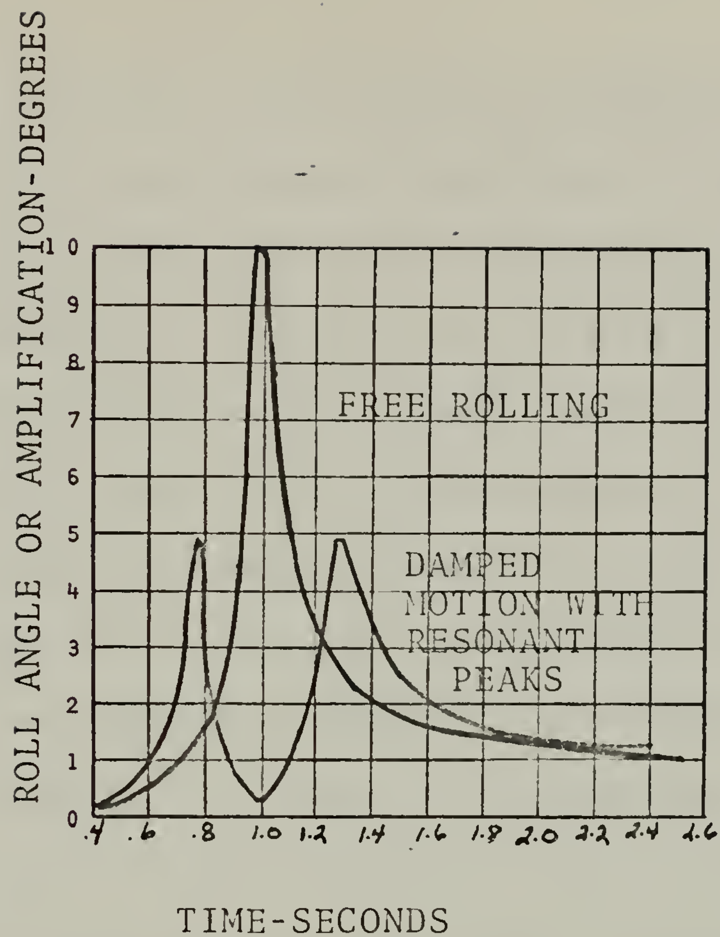

Figure 9. Calculated Characteristic Curve for Passive Tank Stabilizer.

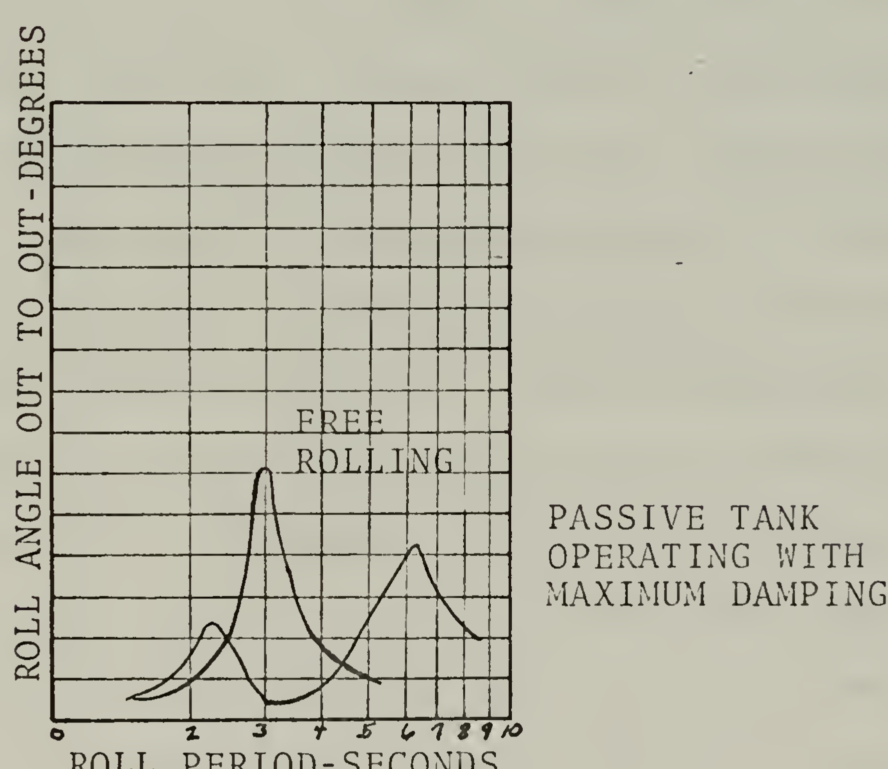

Figure 10. Characteristic Curves for a Passive Tank Stabilizer Disturbing Force ± 0.8° Wave-Slope.

relate to the simplest U-tube type of stabilizer and in certain cases the prediction of the motion of the fluid in the tank has been obtained with a certain degree of accuracy but nevertheless the calculations for the relatively simple U-tube arrangement with these considerations are quite complicated, so that it is necessary to introduce further approximations which change the results of uncertain values. It must be pointed out however that the damping of the system is an important factor in practice, because if the internal damping for example was very low, the amplitude of motion of the fluid in the tanks would continue to grow so long as the external sea disturbance was present. Therefore if the free volume in the top of the tanks is not sufficient to accomodate very large oscillations in the fluid level, then its motion would be suddenly restricted and the tanks are no longer effective. However with a high damping the amplitude of oscillation of the fluid will stabilize at a moderate value such that the energy dissipated by internal damping is exactly equal to the energy supplied to the system by the external wave disturbance. From the above considerations it must be pointed out then, that the fluid in a passive tank stabilizer, supplies a means for the transfer of weight from one side of the ship to the other and also provides a means to dissipate energy since the kinetic energy acquired by the fluid when flowing across the tanks must be dissipated before it starts to flow in the opposite direction, hence these two independent factors (weight transfer

and energy dissipation) must be considered when the system is designed.

The efficiency of a passive tank stabilizer is a measure of the reduction or increase of motion in a vessel when the transfer of fluid takes place within the system.

In order to determine the available stabilizing moment from a fluid tank stabilizer, it is necessary to calculate the total weight of fluid which can be moved from side to side of the ship through the channel and multiply this by the effective radius at which the calculated weight acts. Thence there will be a maximum momentum that depends on the weight and the space available in a given application, also the maximum loss metacentric height, $\overline{GM}$, permissible has influence in the momentum because that is a factor that governs the maximum area of the tank. According to the exposed theory the use of passive tanks implies the existence of an "amplification factor," which in favorable cases could be as high as 6. That is, if a ship has a roll of ± 1 degree, the fluid acquires an equivalent movement of 6 degrees. Practical values of amplification factors are in general around 3. In Reference 5 are some examples showing various systems with different aplification factors.

D. LINEAR MATHEMATICAL MODEL FOR PASSIVE TANK STABILIZER

A mathematical model can be formulated which could be based on the theory of the damped vibration absorber already mentioned, combined with the use of the equation of motion.

In order to formulate the model for the combined motion of a ship with a tank stabilizer two differential equations are going to be used, one that takes care of the tank behavior and the other one that takes care of the ship behavior. The equations are presented as given by G. J. Goodrich [8], and are, for the tank:

$$\ddot{\theta} + k\dot{\theta} + w_t^2\theta = w_t^2\phi - w_t^2 S / g\ddot{\phi} \qquad \text{(III.1)}$$

and for the ship:

$$\ddot{\phi} + K\dot{\phi} + w_s^2\phi - \mu w_s^2\theta = w_s^2 a \cos(wt) \qquad \text{(III.2)}$$

where:

θ = The effective slope of the water in the tank relative to the ship (see Figure 11)

$\dot{\theta}$ = The angular velocity of the water in the tank

$\ddot{\theta}$ = The angular acceleration of the water in the tank

k = The tank damping coefficient

w_t = The natural circular frequency of oscillation of the water in the tank

ϕ = The roll angle

$\dot{\phi}$ = The angular velocity of roll

$\ddot{\phi}$ = The angular acceleration of roll

K = The roll damping coefficient

w_s = The natural circular frequency of roll

μ = The free surface correction factor

S = The distance of the tank mean water level above or below the rolling center

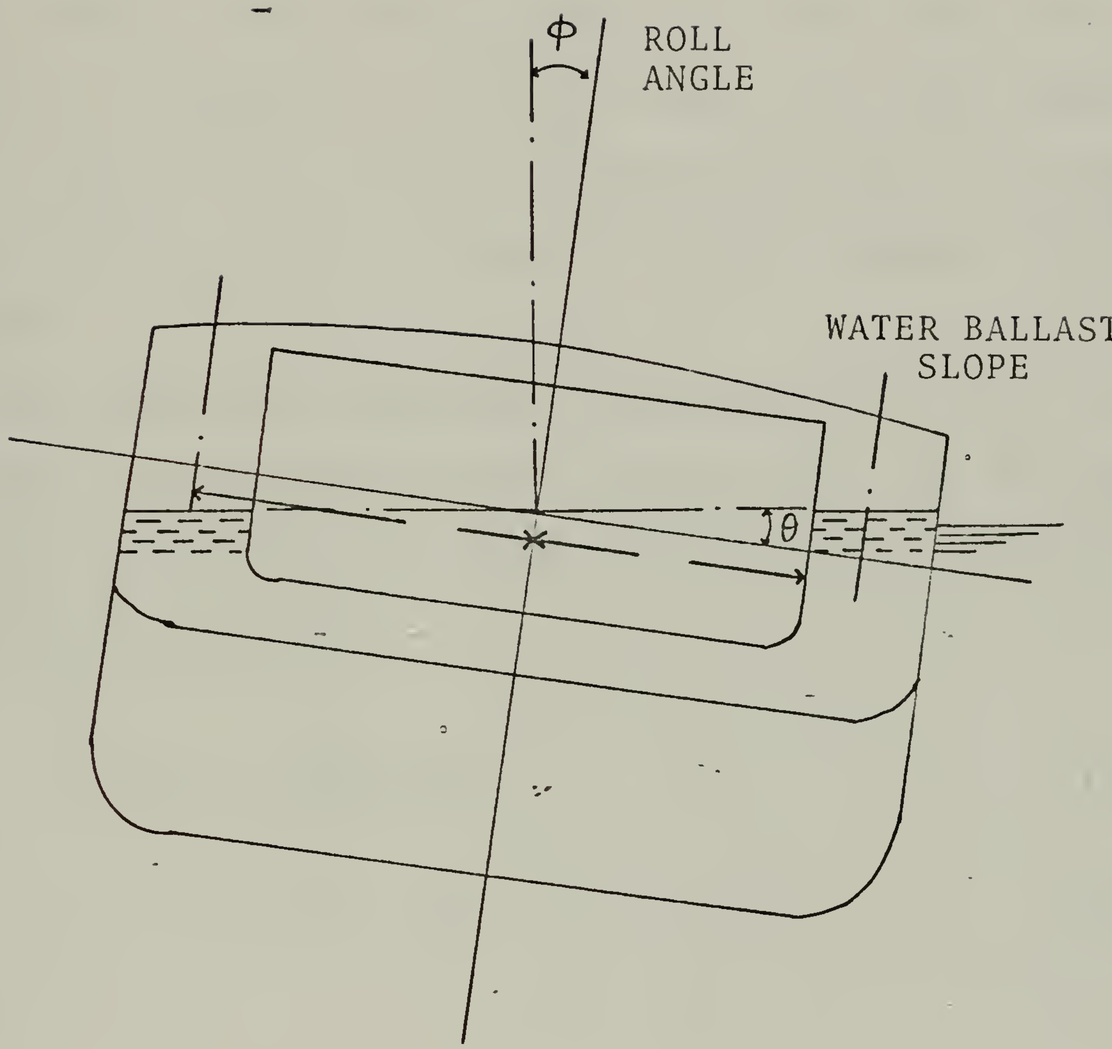

Figure 11. Tank Stabilizer. Cross Section and Coordinates.

a = The wave slope

w = The circular frequency of the forcing function.

Equations III.1 and III.2 give solutions as functions of the different parameters involved. These solutions can be used to investigate the effects when the tank parameters (k, w_t, μ and S) are altered; one way to do this is by using various values of each parameter in turn, keeping the others constant.

The solution of Equations III.1 and III.2 lead to expressions for the amplification factor $\overline{A}$ (w) and the phase angle lag ε of the roll to the forcing function. The expressions are given by

$$\overline{A}\,(w) = \frac{w_s^2}{(X^2+Y^2)^{\frac{1}{2}}} \tag{III.3}$$

$$\varepsilon = \tan^{-1}\frac{Y}{X} \tag{III.4}$$

where:

$$X = (w_s - w\ ) - \frac{\mu w_t^2 w_s^2 (1+Sw^2/g)(w_t^2-w^2)}{(w_t^2-w^2)^2+K^2w^2}$$

and

$$Y = w\left[K + \frac{\mu k w_t^2 w_s^2 (1+Sw^2/g)}{(w_t^2-w^2)^2+K^2w^2}\right].$$

Other factors involved in this specific problem are: the phase lag ε' of the water motion to the ship motion i.e.

$$\varepsilon' = \tan^{-1} \frac{k_w}{(w_t^2 - w^2)} \tag{III.5}$$

For the static case $\theta = \phi$ and the static moment:

$$M' = w_s^2(1-\mu). \tag{III.6}$$

From the above static moment, the percentage loss of roll stiffness F as a result of the free surface effect of the tank will be:

$$F = 100\mu. \tag{III.7}$$

Observe that the two differential equations for the tank and the ship are cross coupled in θ, ϕ, and $\ddot{\phi}$ and due to the angular acceleration of the roll, the quantity $-w_t^2 S\ddot{\phi}/g$ accounts for the effect of the horizontal acceleration of the water in the tank.

In conclusion the equations formulated here, give only a qualitative indication of the effects of the variation in the tank parameters.

Figure 12, shows the roll response curves obtained from experiments on a model in which the tank size varied, and Figure 13 shows the calculated roll response curves corresponding to the results shown in Figure 12 [8].

E. ACTIVE TANK STABILIZERS

The requirements for better stabilization than that provided by passive stabilizers has arisen in some applications. Oceanographic research vessels and ballistic missile-launching surface ships are examples of ships requiring the

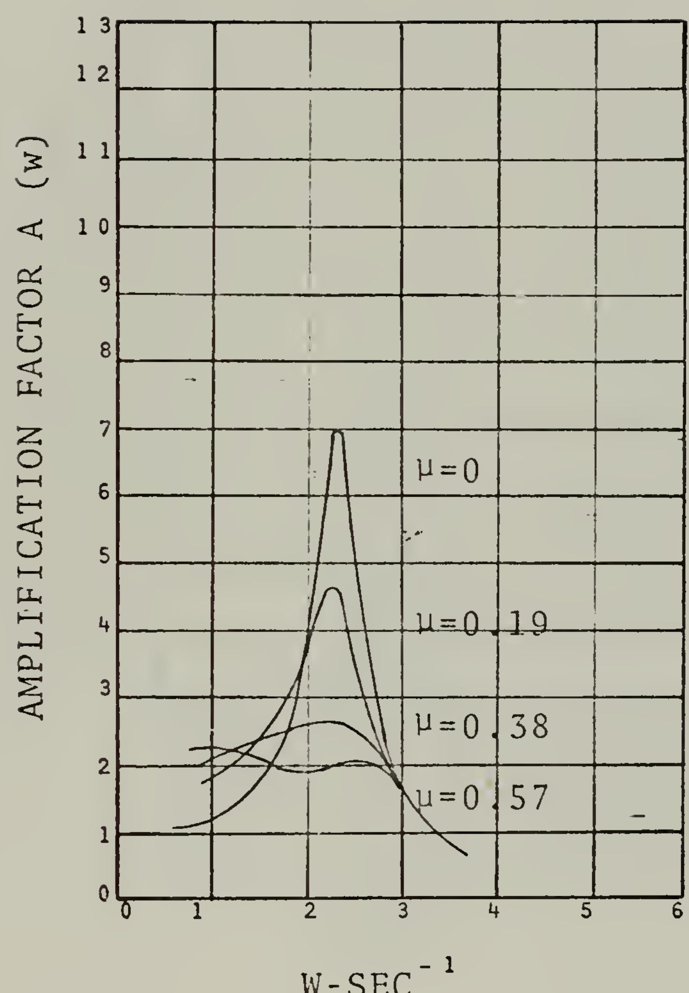

Figure 12. Experimental Curve for an Active Tank Stabilizer.

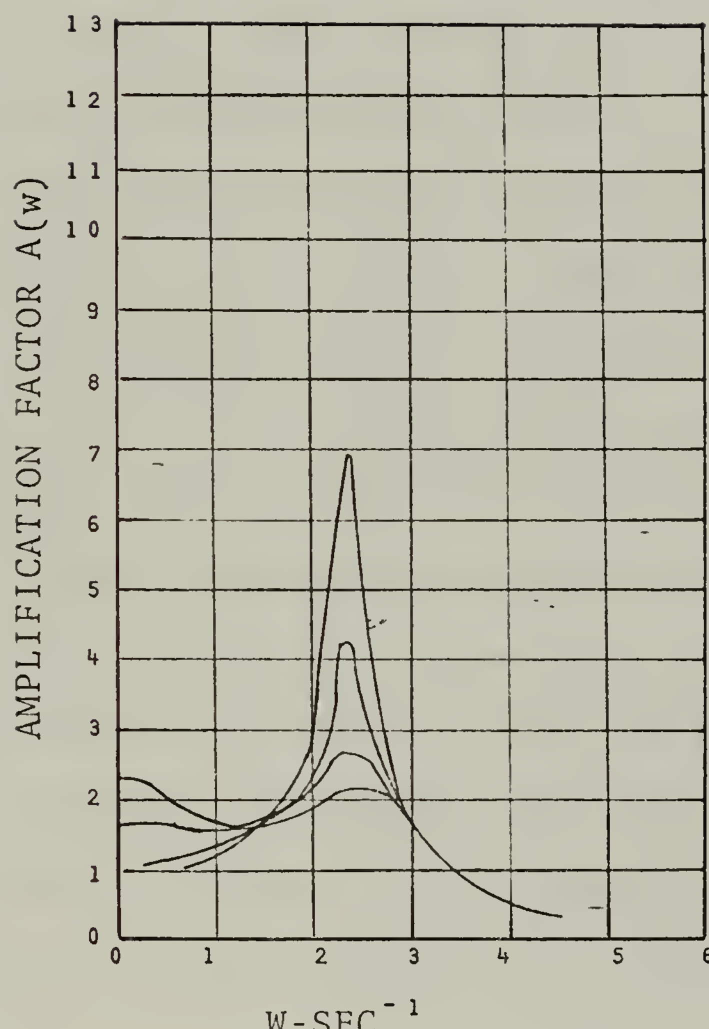

Figure 13. Calculated Curve for an Active Tank Stabilizer.

best possible antiroll system. They operate at very low speeds and in some cases at zero speed and under these conditions the use of stabilizers such as fins is impractical. Active tank stabilizers are well suited to provide the increased stabilization effectiveness [9].

One of the most typical arrangements for active tank stabilizers is the one shown in Figure 14. The system consists of two tanks interconnected through a channel. A water pump is introduced into the connecting canal, whose speed and direction of rotation are maintained constant and the only varying parameter is the propeller pitch of the pump, and this variation provides the approach for realizing a rigid regulation of the stabilizing moment. In Figure 15 are shown experimental results obtained with such pumps and gives a practically linear relation between the angular position of the blades, β, and the instantaneous discharge Q. From this is demonstrated that the angular velocity of flow of the water in the tanks, $\dot{\theta}$, is linearly dependent upon the angular position of the blades [3], that is:

$$\dot{\theta} = k\beta \qquad \text{(III.8)}$$

where k is a coefficient of proportionality.

The displacement θ of the water tank surface can be determined assuming that the pitch β is a function of time, i.e., $\beta = \beta(t)$, therefore:

$$\theta = k\int_{t} \beta(t)dt. \qquad \text{(III.9)}$$

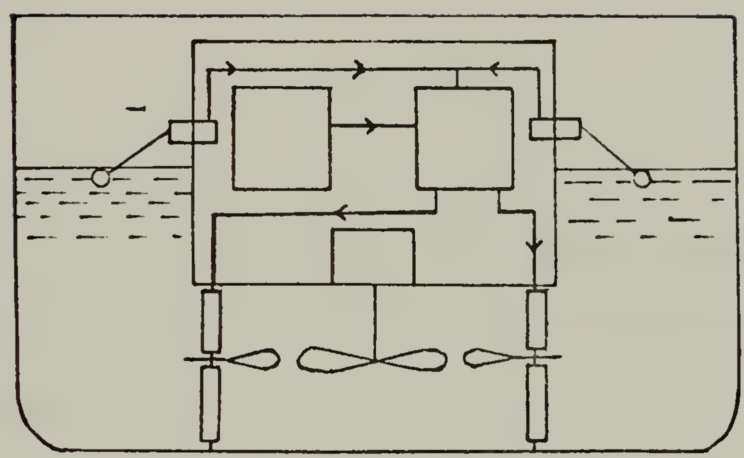

Figure 14. Active Tank Stabilizer with Water Pump.

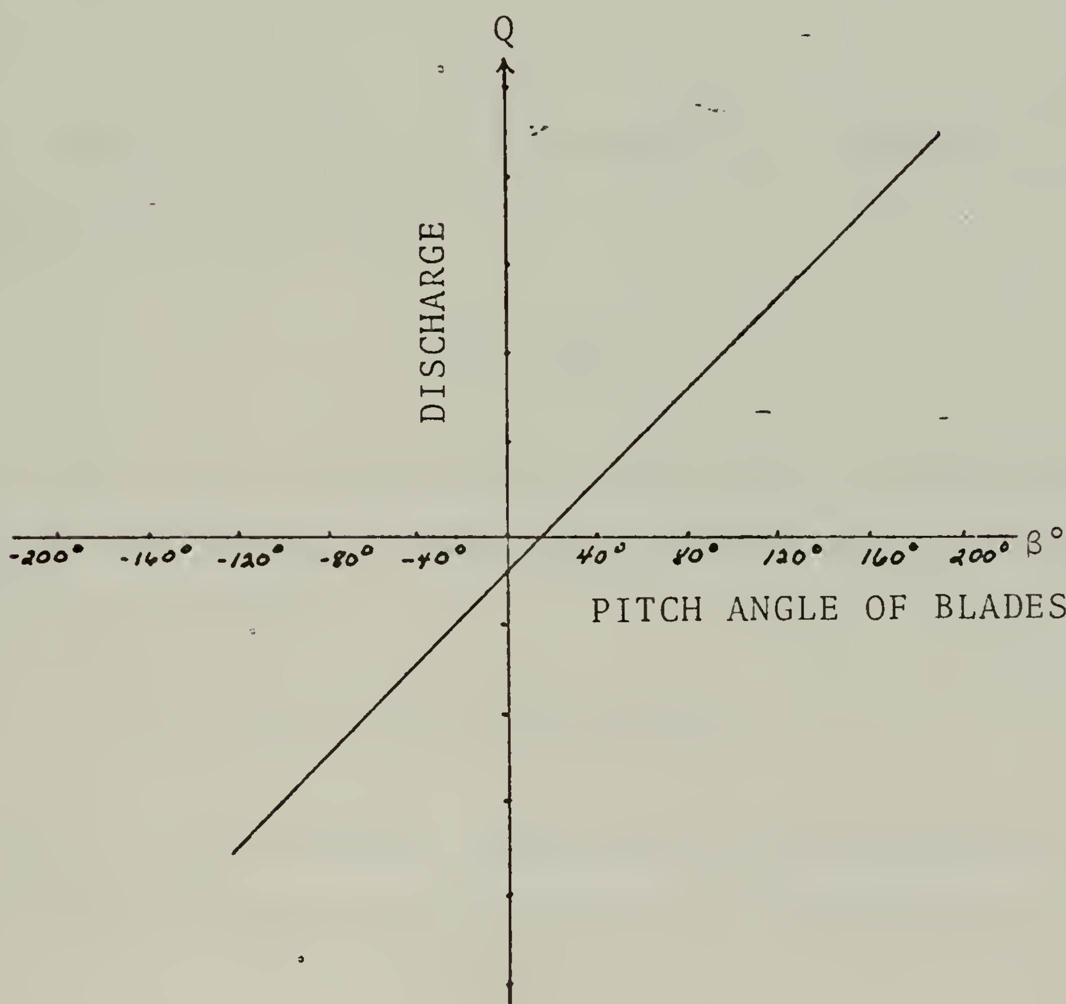

Figure 15. Linear Relationship Between β and Q.

The above assumption also provides a law that governs the changes in the stabilizing moment hence:

$$M_s = 2\rho F_o c\theta \tag{III.10}$$

where F_o is the area of the free surface of the fluid level in the tank and c is the distance from center of gravity of the ship to the center of the cross section of the tank as shown in Figure 11.

Examining the effect of the stabilizing tanks on the assumption that the stabilizing moment is proportional to $\dot{\phi}$, the angular velocity of roll, and opposite in sign then the equality

$$\theta = -h\dot{\phi} \tag{III.11}$$

holds, and on the basis of Equation III.8, gives the following expression for pitch, β:

$$\beta = -\frac{h}{k}\ddot{\phi} \tag{III.12}$$

where h and k are constants of regulation. According to this, the differential equation of roll motion of the stabilized ship becomes:

$$\ddot{\phi} + 2\nu_s\dot{\phi} + w_s^2\phi = w_s^2 a \sin wt \tag{III.13}$$

where $2\nu_s$ is the reduced coefficient of resistance to the motions of the stabilized ship and it is expressed as:

$$2\nu_s = \frac{2N + 2\ c\ F_o\ k}{I + \delta I}\ . \tag{III.14}$$

F. CONCLUSIONS

Active tanks are more effective than passive tanks because with a proper choice of characteristics, they stabilize the vessel with any combined values for natural period of ship motions and wave period.

Passive tanks are truly effective when operated in conditions at or near resonance.

It has been demonstrated that in an irregular sea the active tank configuration is more effective too.

If the tanks are sufficiently high, the introduction of an activated system is effective because it will increase the swings of the oscillations of the fluid level within them as compared to the passive types.

In an activated system the natural oscillations of the liquid are specified independently of the period of oscillation of the ship, thus permitting the dimensions of the water canal of the tanks to be reduced in comparison with those of passive tanks, and this results in some reduction in the weight and space occupied by the system.

If the vessel has a constant heel due to assymmetry of the loading or any other cause, then under these conditions the operation with active tanks is more effective than with passive tanks.

It is possible to generate ship motions in a calm sea with the help of active tank stabilizers, this is useful in some type of ships for some applications, for example in ice-breaker ships in which the generated motion can help to break the ice.

There are some other advantages in using active tanks, but it is also true that they have some basic disadvantages such as their complexity, high cost of installation, the requirement of extra energy to be operated, the use of some complex control device, and the extra cost of maintenance.

IV. GYROSCOPIC STABILIZERS

A. GENERAL

The second kind of stabilizer to be discussed in this thesis is one whose stabilitizing moment is generated by heavy gyroscopes suitably installed on board the vessel. Actually this system is generally used in ships whose operating conditions are restrained to the range of low or zero speed, as is the case, for example, in oceanographic research vessels. It is at those operating conditions that rolling is generally most severe, and the stabilization of roll is expected to have a significant improvement in operational efficiency.

Gyroscopic stabilizers have been used since 1904 when a passive type of syrostabilizer was invented in Germany by Schlick [12]. Seven years later appeared installation of the active type, furthermore in the United States there were some other installations by Sperry and others. One of the best known gyrostabilizer installation was mounted on board the Italian passenger ship SS Conte di Savoia, the installation consisted of three large rotating gyro wheels.

B. DISCUSSION

The use of gyroscopes as stabilizers of rolling motion is based on the property of a fast rotating gyroscope to resist, under certain conditions, changes in the spatial direction of its axis of rotation.

Consider a gyroscope having three degrees of freedom, as shown in Figure 16, if a pair of forces F is applied to the external ring of the gyroscope, its axis will tend to rotate in a plane perpendicular to the plane determined by the lines of action of the pair of forces, and the direction of rotation is such, that the vector of the angular velocity of rotation of the flywheel will move in the shortest way to coincide with the vector of the external moment, that is, the axis of the gyroscope acquires precesssion motion and the angular velocity of this precession is indicated as $\dot{\psi}$ in Figure 16. The result is that the plane of the gyro wheel moves coinciding with the plane of action of the external pair F.

If the gyroscope is constrained in such a way that it has only two degrees of freedom, then when the gyroscope is revolving about a perpendicular axis lying in the plane of the gyro wheel, a gyroscopic moment is created which acts upon these constraints, see Figure 17.

It must be observed that the precession of the gyroscope in the constrained case induces a reactive moment which acts upon the constraints applied to the gyroscope.

In Figure 18 can be observed that the symmetrical flywheel of a gyroscope is allowed to rotate about an axis AA' with an angular velocity w_1, the axis AA' attains, in turn, an angular velocity w_2 about one of the principal axes of inertia of the gyrowheel. In general w_1 is much larger in comparison with w_2, and in these conditions may be considered

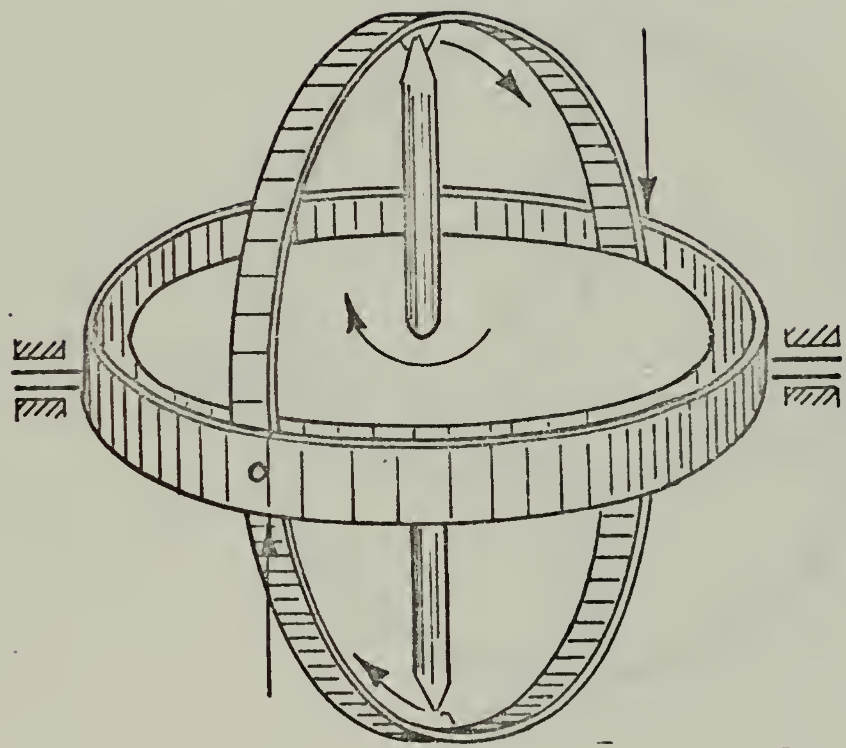

Figure 16. Forces and velocities Acting on a Gyroscope.

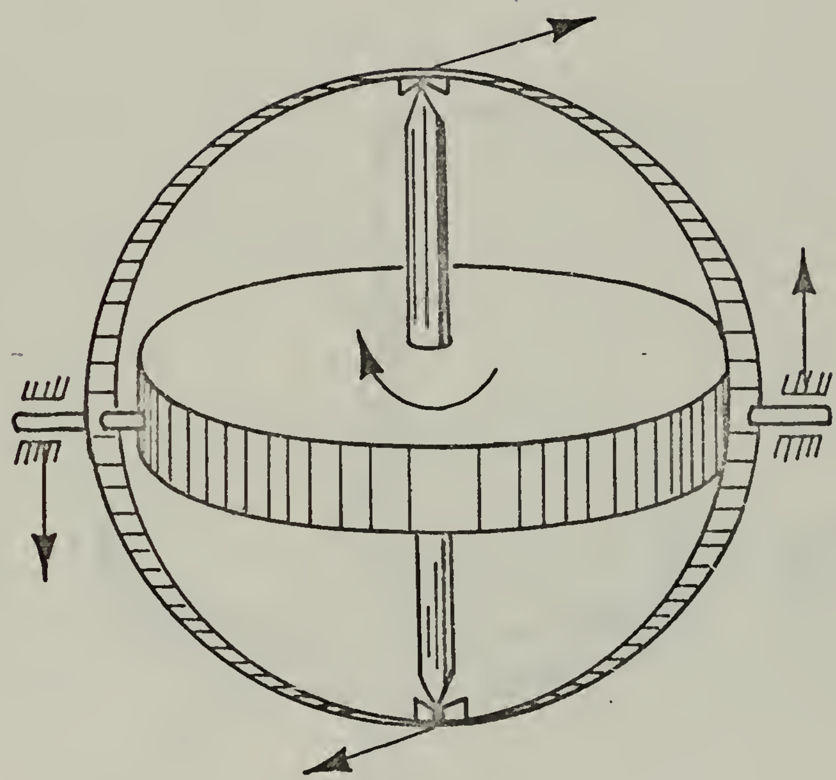

Figure 17. Constrained Gyroscope.

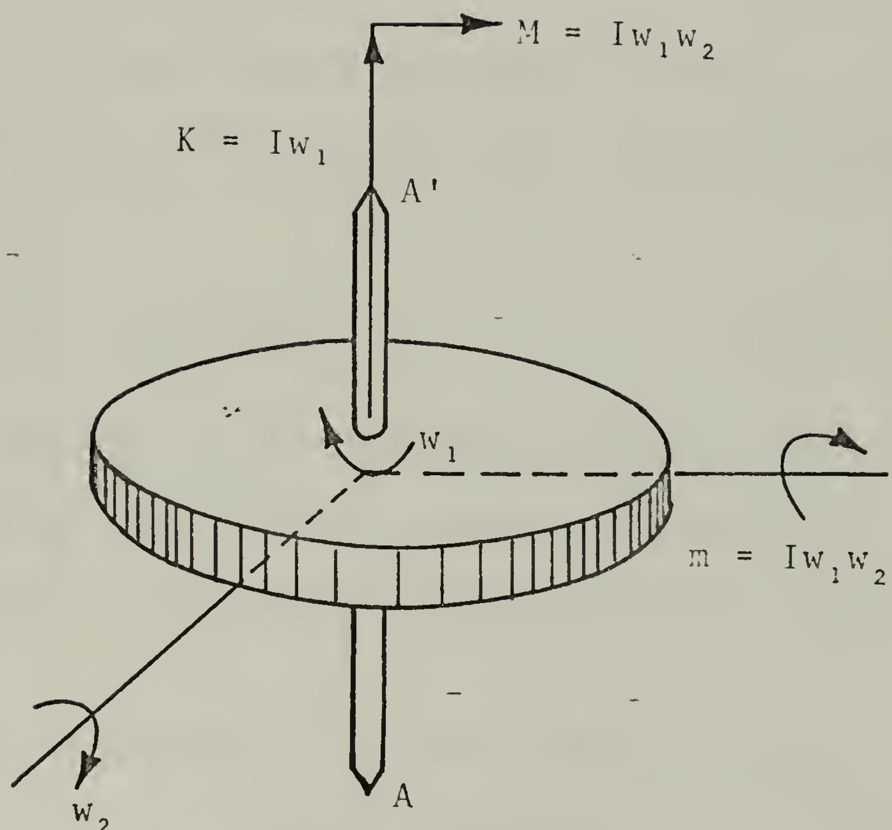

Figure 18. Moments Acting on a Rotating Gyroscope.

that the vector k of the principal moment of momentum of the gyroscope is directed along the axis of rotation AA', this moment is equal to Iw_1, I being the moment of inertia of mass of the flywheel with respect to the axis AA'. According to the angular momentum law [3], the moment M, of the external forces acting upon the gyroscope and inducing precession w_2 will be:

$$M = kw_2 = Iw_1w_2 . \tag{IV.1}$$

In the case of the constrained gyroscope shown in Figure 17, the gyroscopic moment acting upon the constraints will be:

$$M_1 = k\dot{\psi} = Iw_1\dot{\psi} . \tag{IV.2}$$

Directions of precession and gyroscopic moments are determined according to the following corkscrew principle indicated in Reference 3, (see Figure 19) and written here for reference:

1. If a corkscrew, directed in the sense opposite to that of the moment vector of the external forces, is turned through 90 degrees in the direction of rotation of the gyrowheel, then its rotation will indicate the direction of the precession.

2. If the corkscrew, directed in the sense opposite to that of the vector of the angular velocity of precession, is now turned by 90 degrees in the direction of rotation of the flywheel of the gyroscope, then its own rotation will indicate the direction of the gyroscopic moment.

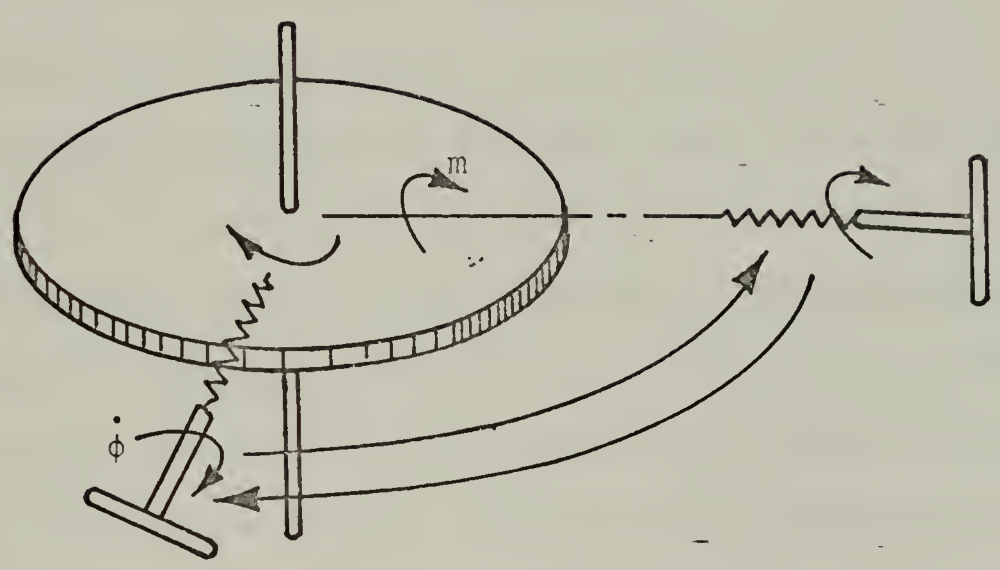

Figure 19. Corkscrew Principle

These two directions may also be determined by means of the rule indicating that the wheel always tends to rotate its own plane following the shortest route into the plane of the external couple F, or to force its own axis of rotation to coincide with the precession vector.

The planes of rotation of the gyro wheel, of external moment and of precession are mutually perpendicular.

From this it is clear that the foundation for gyroscopic stabilization is based on the principle of conservation of momentum, and this concept will help in the following description of a gyroscopic stabilizer.

A gyroscopic stabilizer consists basically of a massive gyroscope installed in a frame, and it can rotate in trunnions fixed rigidly to the hull of the vessel. Figure 20 is an illustration of such a system, the gyroscope has two degrees of freedom defined as the rotation of the gyrowheel and the turning of the frame in the plane of symmetry. The third degree of freedom, that is, the rotation of the frame in the plane of the section of the ship, is limited by the hull of the vessel and for this reason the gyroscope appears to be constrained. The ship itself constitutes the external ring of a free gyroscope in a gyroscopic stabilizer.

As in tank stabilizers, the gyroscopic stabilizers can be distinguished as active or passive. In most of the passive type gyrostabilizers, precession arises as a reaction to the motions, while in the active type, precession is the result of the action of some special activator whose operation is regulated by means of some automatic controller.

Referring to Figure 20, the vessel rolls with an angular velocity $\dot{\phi}$ which at a certain instant of time is in a clockwise direction. As the gyroscope turns with the angular velocity of the motions, it generates a gyroscopic moment whose action upon the gyroscope's frame induces a precession in the plane of the vessel and with a velocity $\dot{\psi}$. According to the corkscrew principle mentioned before, the vector $\dot{\psi}$ must be directed to port side.

A gyroscopic moment is produced due to precession in the plane of symmetry, the moment is applied to the trunnions and acts in the plane of the framework. The projection of this moment on the plane of the ship's frame gives the required stabilizing moment M_s, and is expressed as follows:

$$M_s = j\Omega\dot{\psi} \cos\psi \tag{IV.3}$$

where

j = Moment of inertia of mass of the gyro wheel about its axis of rotation.

Ω = Angular velocity of rotation of the gyro wheel.

Similar to the case in tank stabilizers, the direction of the moment M_s is opposite to the angular velocity $\dot{\phi}$, see Figure 20, and the gyroscope moment that originates precession of the frame is given by:

$$m = j\Omega\dot{\phi} \cos\psi \ . \tag{IV.4}$$

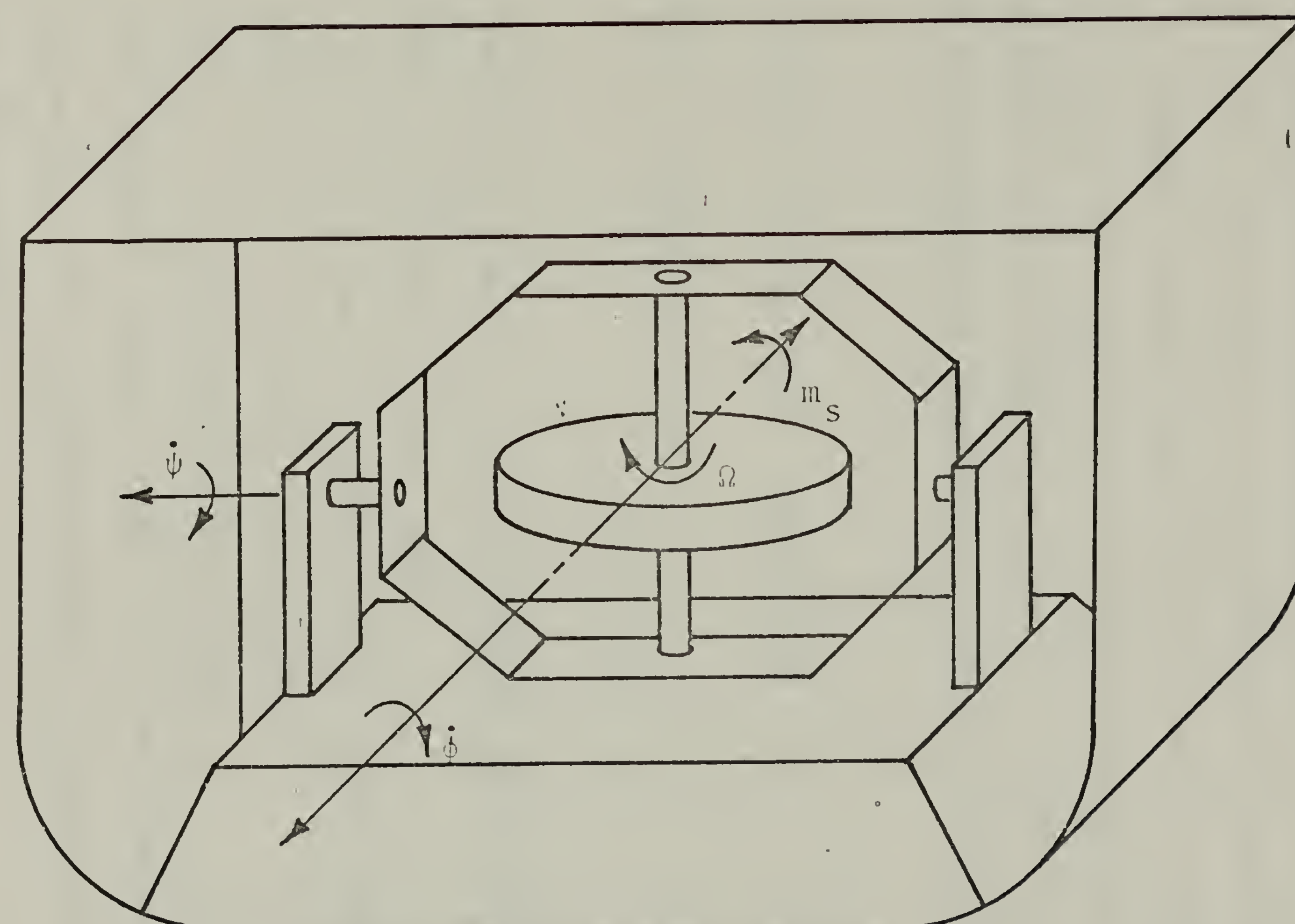

Figure 20. Gyroscopic Stabilitzer.

C. LINEAR MATHEMATICAL MODEL FOR GYROSTABILIZERS

A mathematical model for a gyroscopic stabilizer can be determined with the formulation of the differential equations of the motions of the ship and the differential equations for the precession of the stabilizer gyroscope's frame. It is assumed in this formulation that the oscillations of the vessel are small and the orbital motion of the center of gravity of the ship is neglected. Under these assumptions the formulated equations as given in Reference 3 are:

$$(I+\delta I)\ddot{\phi} + 2N\dot{\phi} + WGM\phi = WGMa\sin wt - M_s \qquad \text{(IV.5)}$$

$$T\ddot{\psi} = - H\sin\psi + m + \varepsilon F \qquad \text{(IV.6)}$$

where:

T = Moment of inertia of the gyroscope about the axis of rolling of the frame.

H = The static moment of the rolling system about the axis of rolling of the frame.

ψ = Angle of precession of the frame.

εF = The additive moment, applied to the frame of the gyroscope, this value is chosen at the discretion of the designer.

The applied reactive moment εF does not affect the rolling motion because it is applied to act in the plane of symmetry of the vessel, this quantity may follow some law or may be taken equal to zero.

Introducing in Equation IV.5 and in Equation IV.6 the values of M_s and m obtained in Equations IV.3 and IV.4 respectively, the following equations are obtained:

$$(I+\delta I)\ddot{\phi} + 2N\dot{\phi} + WGM\phi = WGMa \sin wt \qquad \text{(IV.7)}$$

$$T\ddot{\psi} + H\sin\psi = j\Omega\dot{\phi} \cos\psi + \varepsilon F. \qquad \text{(IV.8)}$$

For simplicity the following notation is defined:

J_s = (I+ I) = Total moment of inertia of ship about axis of roll, lb-ft-sec.

B_s = 2N = Hydrodynamic damping coefficient, lb-ft-sec.

K_s = WGM = Static righting moment coefficient.

a_e = = Effective waveslope disturbance, rad.

a_s = = Effective stabilizer waveslope, rad.

Effective waveslope is defined here either as a disturbing or a stabilizing moment about the roll axis, expressed in terms of the static list which this moment will produce.

From the above definition, Equation IV.7 can be expressed as:

$$J_s\ddot{\phi} + B_s\dot{\phi} + K_s\phi = K_s(a_e - a_s). \qquad \text{(IV.9)}$$

In the roll Equation IV.9, various coupling effects become unimportant at low speed since they vary as the square of speed, therefore, they have been neglected.

The action of the gyro as a stabilizer is found by controlling $\dot{\psi}$, the velocity of precession about the athwartship axis. If the spin axis is in vertical position, the reaction moment will be given by:

$$M_t = \dot{\psi}\Omega \frac{T}{g} \text{ lb-ft.} \tag{IV.10}$$

The resultant component of the moment M_t that tends to stabilize the vessel when roll occurs is proportional to the cosine of the precession angle ψ. The equivalent stabilizing waveslope a_s defined before as the stabilizing moment in terms of the static list which it would produce is expressed as:

$$a_s = \frac{\dot{\psi} \cos \psi}{WGM} . \tag{IV.11}$$

Figure 21 is a simplified block diagram of a gyrostabilizer system, observe that the ship responds to a net roll moment which results from the action of both the waves and the gyrostabilizer. Also is indicated in the diagram the fact that even without a control signal, the dynamics of the ship and stabilizer are not independent.

Similar to other systems, three control signals must be considered, they are, roll angle, roll velocity, and roll acceleration.

1. Roll Angle

This signal is not useful for control because the stabilizer has a capacity equal to zero at zero frequency,

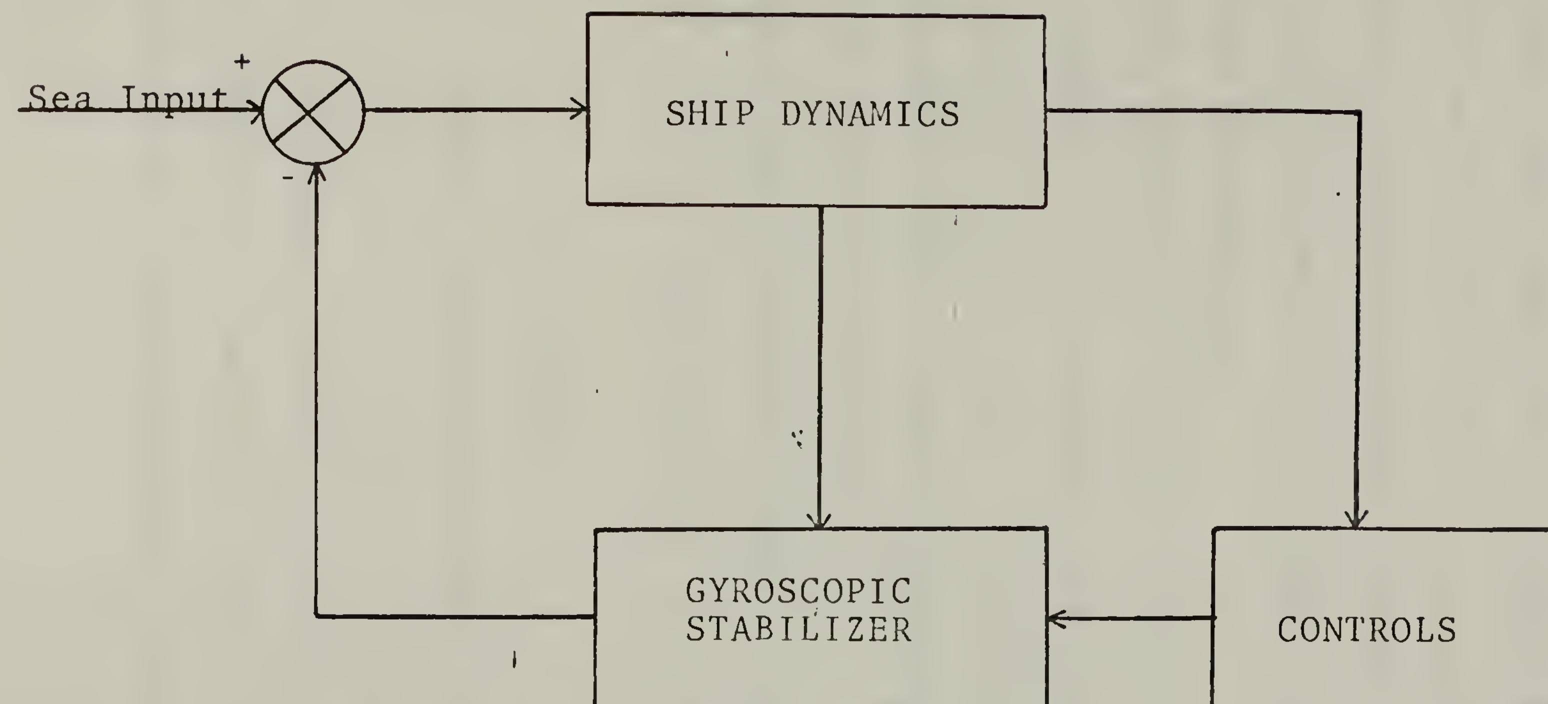

Figure 21. Block Diagram. Gyroscopic Stabilizer.

which means that the stabilizer cannot correct for a static list since this would require it to precess indefinitely in one direction.

2. Roll Velocity

This signal generates a stabilizing moment which is proportional to the roll velocity and would serve in the same sense as the ship's natural damping. Roll velocity, as in the other systems, is the primary control signal.

3. Roll Acceleration

This signal could be used in addition to the roll velocity signal if it is necessary to improve the response to high frequency disturbances. A gyroscopic stabilizer has an inherently rapid response therefore this control signal would not be too important.

After the above considerations, a simplified control equation, to a first approximation will be:

$$\psi = \int k \dot{\phi} \, dt \tag{IV.12}$$

where k is a control sensitivity coefficient, (ordered - precession-velocity/roll velocity), similarly $k\dot{\phi} = \dot{\psi}$, i.e., the ordered precession velocity in rad/sec.

It is pointed out that some limits exist on the precession velocity and precession angle indicated in Equation IV.12.

D. CONCLUSIONS

The main advantage in installing an active gyrostabilizer, is its ability to reduce the rolling action effectively

in any sea condition and with different ship loading conditions, on the other hand they are also efficient if the abruptness of the motions increase, which is to say that a decrease in their period occurs.

The basic disadvantages associated with the installation of gyrostabilizers are:

1. Their significant weight.
2. Difficulties of arrangement.
3. Large initial cost.
4. Their tendency to shake the hull connection loose.
5. Complexity of installation and operation.
6. The safety conditions are limited because the rotor of the gyroscope can accumulate within itself a tremendous amount of kinetic energy which in case of an accident could cause great damages on the vessel.

V. ACTIVATED FIN STABILIZERS

A. GENERAL

The most efficient method actually used to reduce the rolling motion is by means of movable fins conveniently installed on board the ship and whose function is to generate the required stabilizing moment. A large number of modern ships are provided with a roll damping system consisting of bilge keels to increase passive damping, plus servo-controlled fins in pairs, whose motion is governed by some instrumentation such as gyroscopes that sense the ship's rolling.

The fins act like small aircraft wings developing hydodynamic lift and originating a moment that opposes the ship's roll. They are very effective where stabilization is wanted at speed, thus they are chosen in a wide variety of commercial and Navy ships. The British have been the pioneers in the fin stabilizers field and have outfitted a large number of ships including the "Queen Mary." Various means of control have been developed and in References 2, 4, and 11 are detailed the evaluation and analysis of some of them.

The United States also has developed some stabilizer systems using fins, in 1955 the Sperry "Gyrofin," was developed. It operated on the same general principles but with some differences in the arrangements for retraction and in the control subsystem. Similar to the other systems,

the Sperry Gyrofin utilizes a rate gyro to provide the primary control signal, but the feedback uses a true vertical gyro reference and an apparent vertical reference obtained from a linear accelerometer sensitive in the sway axis.

The majority of fin stabilizers systems employ variable-displacement pumps and ram-operated flapped fins which may be retracted into the hull of the ship. Some installations however have used non-retractable, unflapped fins with servo valves. But in any case the installation of a particular system is limited by size and cost considerations.

B. DISCUSSION

In general, the movable fins consist of controlled side rudders projecting downward and outward through the hull on both sides of the vessel at the turn of the bilge, and may be rotated on stems. Each fin is clamped to a stem whose inboard end is geared in such a way that a pair of stems rotate the same amount in opposite directions. The stems are rotated by means of a special automatically controlled drive.

Figures 22 and 23 show a typical placement of fins on ships, in profile and cross section views respectively.

Refer to Figure 22, consider the ship moving at the velocity v in calm water. It is assumed that the ship is maintained in upright position due to the action of some external couple, so that it is not allowed to roll, aa'

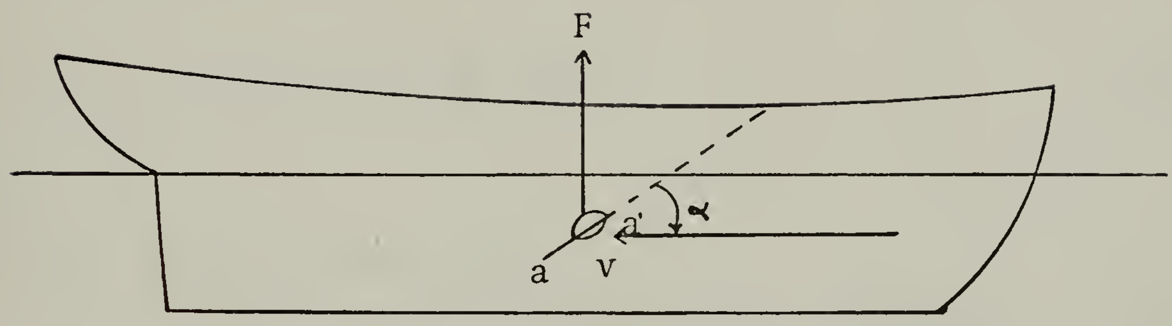

Figure 22. Fin Stabilizer. Side Location.

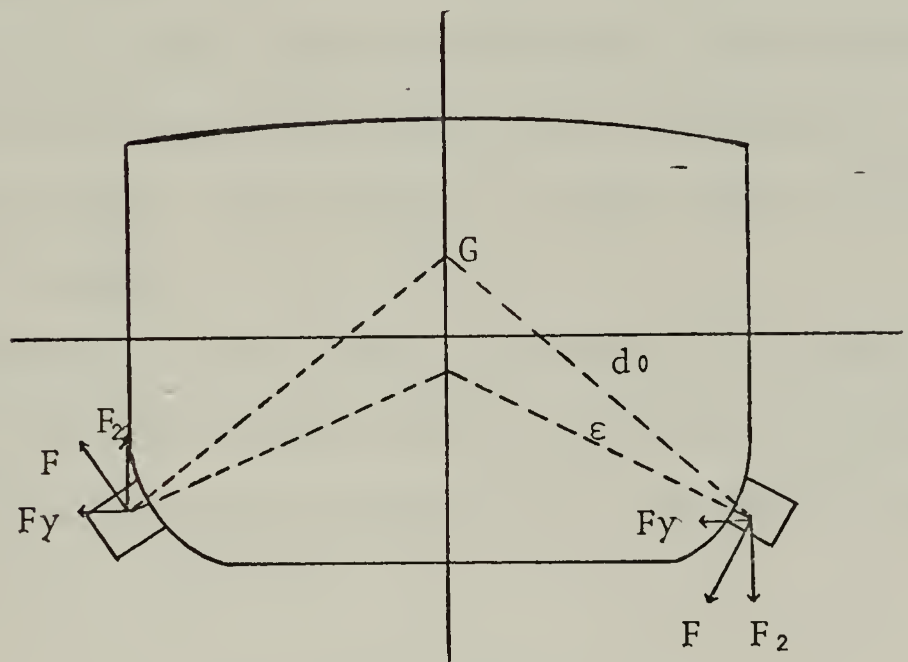

Figure 23. Fin Stabilizers. Cross Section.

represents the starboard fin extended beyond the hull. When the fin is rotated an angle α from its middle position, a lift force is developed upon it, with its direction upward and it is expressed as:

$$F = C_1 \frac{\rho}{2g} A_f v^2 \qquad (V.1)$$

where

A_f = Area of the fin.

v = Velocity of the oncoming flow equaling, in the given case, the forward velocity of the ship.

C_1 = Dimensionless lift coefficient.

In Figure 22 it is observed that the force F must be perpendicular to both the velocity v of the approaching flow and the fin's axis. Suppose now that a symmetrically placed fin in the other side of the ship is inclined also to an angle α, but in the opposite direction, then on that fin is developed the same lift force F but now directed downwards. Thus the action of the force F on each fin as indicated will produce a moment.

In Figure 23 the ship is observed in a front view, showing a particular position of the fins. The moment of both lift forces with respect to G, the center of gravity of the vessel, will be:

$$M = 2Fd_0 \cos \varepsilon \qquad (IV.2)$$

where ε and d_0 are as indicated in Figure 23. The angle ε

is formed by the fin stem and a normal dropped from the center of the fin to the longitudinal axis passing through G. In practice the angle ε is so small that it can be assumed that $d_0 \cos\varepsilon \simeq d_0$.

The coefficient C_l appearing in Equation V.1 has a linear relation with fin angle α, given by

$$C_l = \frac{dC_l}{d\alpha} \alpha. \tag{V.3}$$

Figure 24 shows graphically the linear relationship indicated in Equation V.3.

It must be pointed out that C_l also depends upon the shape of the profile and the relative span.

Defining:

$$C_l' = \frac{dC_l}{d\alpha}$$

and substituting Equations V.1 and V.3 into Equation V.2, an equation for the moment becomes:

$$M = C_l' d_0 \frac{\rho}{g} F v^2. \tag{V.4}$$

Assuming that the fin angle α obeys a harmonic law for its changes, the flow about the fins is no longer steady, and for an accurate analysis of the fin operation, this should be based on the theory of a rotating wing, i.e., the problem will complicate. The unsteadiness effect is then neglected and an approximation is assumed to arrive at a moment equation:

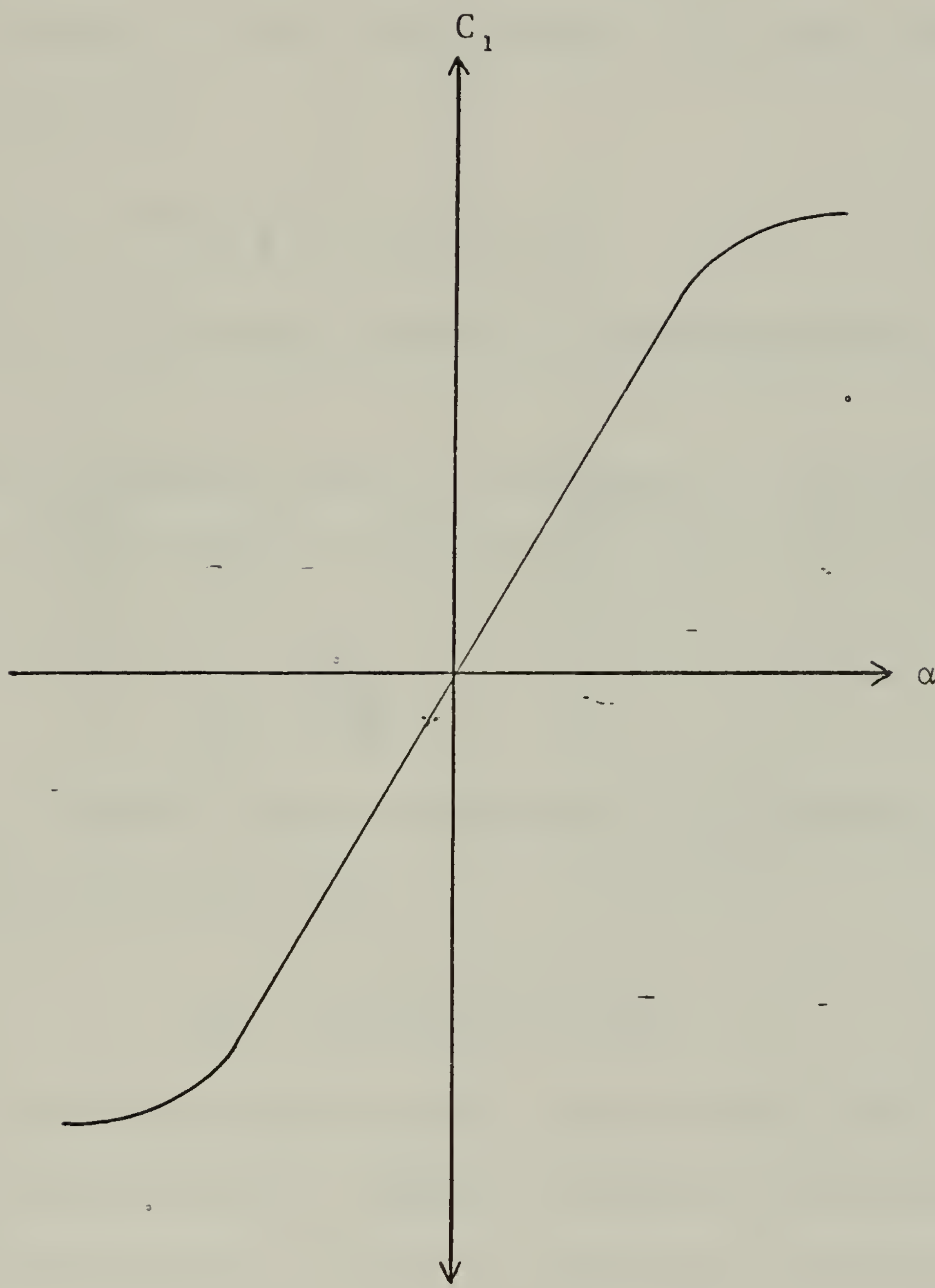

Figure 24. Linear Relationship. Fin Angle α vs. Lift Coefficient.

$$M = C_1' d_0 \frac{\rho}{g} Fv^2 \alpha_0 \sin wt \tag{V.5}$$

where $\alpha_0 \sin wt = \alpha$ is the harmonic law expression.

In this manner it is noted that the generated moment is time dependent and it varies obeying a harmonic law.

For n pairs of fins installed on a ship the stabilizing moment is given by:

$$M_s = knC_1' d_0 \frac{\rho}{g} Fv^2 \alpha_0 \sin wt \tag{V.6}$$

where k is a coefficient related to the unsteadiness of the flow.

Another parameter that is considered in fin stabilizer systems is their static characteristic and it is given by

$$\mu_s = \frac{M_{so}}{GM} \tag{V.7}$$

where M_{so} = maximum stabilizing moment corresponding to the maximum fin angle, i.e.,

$$M_{so} = knC_1' d_0 \frac{\rho}{g} Fv^2 \alpha_0 . \tag{V.8}$$

The equations expressing the moment above, have been derived assuming that the ship is moving in calm water without experiencing rolling. If the ship is subject to the action of waves, then it experiences rolling and in this case the ship motions change the flow pattern about the fins, this change will be taken into account for further analysis.

C. LINEAR MATHEMATICAL MODEL FOR ACTIVE FIN STABILIZERS

In the formulation of a mathematical model suited to analyze the fin-roll-damping system, it is worthy to consider the different aspects from the viewpoint of control engineering. The system is totally activated and basically the scheme represents a large regulator system that comprises:

1. The vessel itself, that is, its response to a roll moment.

2. The roll sensing instrumentation which produces a feedback control signal.

3. The fin dynamics, that is, the relation of fin angle to the roll moment that it produces on the vessel.

With the system as a regulator, it is required from it that the roll angle ϕ, be maintained nearly zero, despite the disturbing sea moment.

Figure 25 is a block diagram representing a fin roll-damping system. Note that the electrical signal output from the instrumentation control is not sufficiently strong, therefore it must be amplified to produce an adequate signal capable of actuating the powerful servomechanism.

In Figure 25 the different blocks are labeled with symbols representing their respective transfer functions to be developed in the following discussion.

1. Ship Dynamics

Assuming that a net moment M_n due to sea disturbances and/or the action of the fins is applied to the ship, then

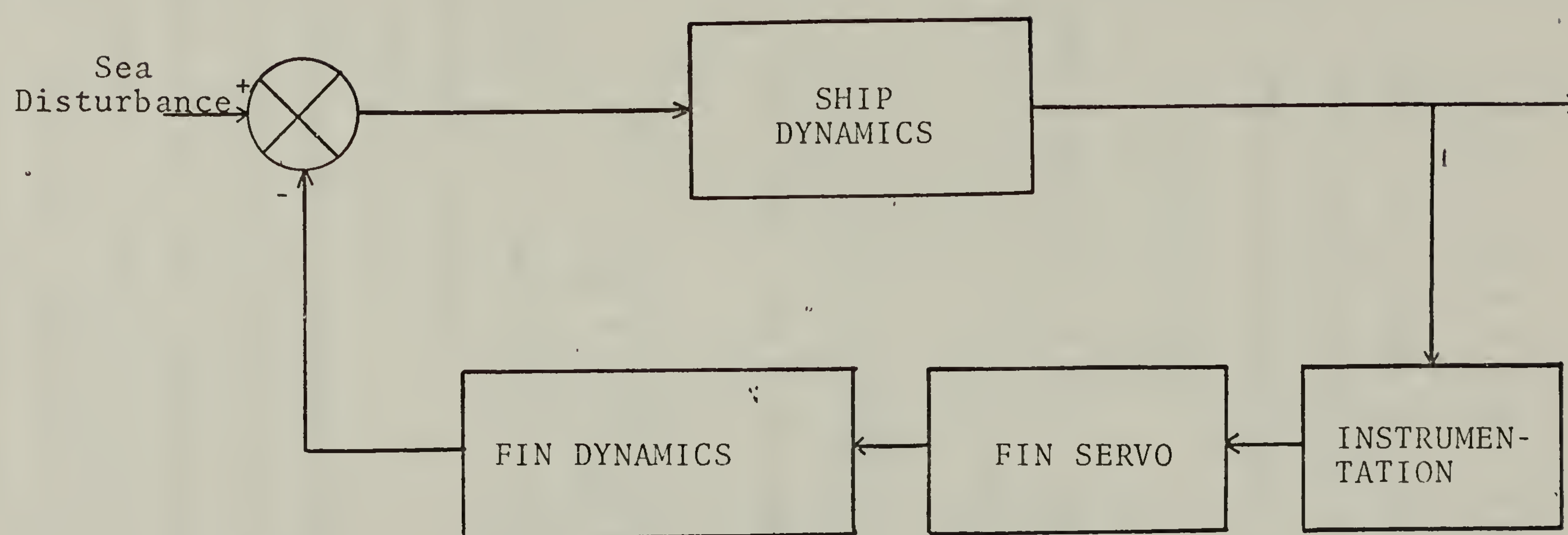

Figure 25. Block Diagram. Fin Stabilizer System.

its dynamic response is given by the following equation of rolling:

$$J_s\ddot{\phi} + B_s\dot{\phi} + K_s\phi = M_n. \tag{V.9}$$

The net roll moment M_n can be expressed also in function of the difference between the disturbing sea moment M_w and the counteracting fin moment, M_f, that is:

$$M_n = M_w - M_f. \tag{V.10}$$

From equation V.9 it is found that the transfer function G_s of the unstabilized ship is given by:

$$G_s = \frac{\phi}{M_w} = \frac{1}{K_s + B_s S + J_s S^2}. \tag{V.11}$$

In quadratic form G_s becomes:

$$G_s = \frac{1}{K_s\left[1 + 2\xi_0 \dfrac{S}{w_0} + \dfrac{S^2}{w_0^2}\right]}$$

where ξ_0 and w_0 are respectively the damping ratio and natural frequency of the ship.

2. Instrumentation

The signal S_f produced by the roll sensing instrumentation is obtained following the same principles mentioned in the study of the other stabilizers systems, i.e., the signal is chosen for primary control which most perfectly complements the dynamic behavior of the ship and stabilizer. It has been demonstrated [13], that in variable-angle activated fins the fin angle should be better controlled by the

roll velocity. However in any case it is desirable to use the three roll signals, i.e., roll angle ϕ, roll velocity $\dot{\phi}$, and roll acceleration $\ddot{\phi}$, with two of them used as auxiliary capacity; from this it is apparent that:

$$S_f = k_1\phi + k_2\dot{\phi} + k_3\ddot{\phi} \tag{V.13}$$

where k_1, k_2, and k_3 are the coefficients of sensitivity to roll angle, roll velocity and roll acceleration respectively.

The transfer function H obtained from Equation V.13 is given by

$$H = \frac{S_f}{\phi} = k_1 + k_2S + k_3S^2. \tag{V.14}$$

The lower and upper "break frequency" in transfer function H are given by: k_1/k_2 for the lower and k_2/k_3 for the upper.

The device used in most stabilizer systems for measuring the roll rate is in general a single-degree-of-freedom gyro.

3. Fin Servomechanism

The fin servo produces a fin angle α which is proportional to the signal S_f. Basically it consists of a hydraulic motor that produces a rate of change of fin angle proportional to the servo error signal; in this case the open loop transfer function has a single integration [14], and is given by:

$$G_\alpha = \frac{\alpha}{S_f} = \frac{k_\alpha}{1 + \frac{S}{w_\alpha}} \tag{V.15}$$

where

w_α = Crossover frequency at which G_α falls to unity generally $w_\alpha >> w_0$.

k_α = Ratio of fin angle to fin command at zero frequency; radians/volts.

4. Fin Dynamics

In the block diagram shown in Figure 25, the fin dynamics is represented by the transfer function G_f, which is multiplied by the fin angle α to produce the fin's roll moment M_f and is given by:

$$G_f = \frac{M_f}{\alpha} = \frac{k_f}{1 + \tau_f S} \tag{V.16}$$

where k_f is a factor that varies with the square of the vessel's speed and τ_f is a small unknown time constant.

5. Stabilized Ship Transfer Function G

From the equations expressing the transfer functions G_s, H, G_α and G_f and Equation V.10, it is found that the transfer function G for the stabilized ship is given by:

$$G = \frac{\phi}{M_w} = \frac{G_s}{1 + G_s H G_\alpha G_f} \tag{V.17}$$

$$= \frac{G_s}{1 + Y} .$$

It is apparent that the loop ratio is $Y = G_s H G_\alpha G_f$.

Equation V.17 can be expressed in terms of the three control coefficients k_1, k_2, and k_3, neglecting any lag that might be present in the fin servo and the hypothetical

time constant τ_f in the fin dynamics, thus the transfer function G is reduced to:

$$G = \frac{1}{(K_s+k_1k_\alpha k_f) + (B_s+k_2k_\alpha k_f)S + (J_s+k_3k_\alpha k_f)S^2}. \quad (V.18)$$

To this degree the effects of the three control terms are to independently increase the effective moment of inertia, damping and restoring moment, and this is explained examining Equation V.18 observe that positive roll angle control increases the apparent metacentric height of the ship, roll rate control however increases the apparent damping and that roll-acceleration control increases the apparent inertia. As it was pointed out before, roll rate is the primary control for fins since damping is what is required. On the other hand roll-angle control improves performance against the lower frequencies encountered in following seas, whereas roll-acceleration control improves performance against the higher frequencies and also contribute to compensate for any lag in the positioning system. Finally it is observed that for given values of k_1, k_2 and k_3, the response of the stabilizer is proportional to the square of the speed. In any system that is required to operate over a significant range of speeds this strong variation must be eliminated by some means of compensation, maybe this compensation can be achieved by adjusting the control sensitivities with speed, according to an inverse square law.

D. CONTROLLED FIN SERVOMECHANISM

The servomechanism subsystem comprise the positioning motor which consists basically of the fin-tilting servomotor and its associated power amplifiers. The basic requirements for an acceptable positioning motor are: minimum use of space, weight, and power, and must meet some positioning moment requirements.

Beside the mentioned requirements, the servomechanism must meet also certain conditions that are imposed by the fin itself, these are, requirements on maximum moment output, maximum velocity of tilt, and maximum acceleration of tilt. These load requirements can be obtained by working backwards, i.e., from system capacity to fin lift, from fin lift to angle of attack, and from angle of attack to positioning moments and angular rates. Finally the servomechanism should meet all these requirements with a motor-to-load power ratio as close to unity as possible, and its response must be sufficiently rapid and accurate.

There are available, certain types of servomotors that can perform their operation satisfying the mentioned demands. These can be:

1. Mechanical Clutches.
2. Electrical Clutches ov varies type.
3. Pneumatic Motors.
4. Electric Motors.
5. Hydraulic Motors.
6. Hydraulic Rams.

Mechanical Clutches, however, are not convenient when smooth continuous control is required. Magnetic clutches on the other hand present some heat dissipation problems when used at certain power. Pneumatic motors are suitable for low power operation where weight must be considered as a limiting factor.

The other three types, electric and hydraulic motors and hydraulic rams are the most used and the most effective types of servomotors. The electric and hydraulic motors present some disadvantages associated with weight, space and simplicity considerations, in addition to this, they must be connected to the fin shaft through a reduction gearing with some stepdown ratio. This constraint not only calls for large expensive gears, but it introduces a potential source of considerable backlash into the system. Thus the most recommended type of servomotor is the hydraulic ram type which does not require a reduction gearing system, therefore the problems associated with the weight, expense, and complexity of reduction gears is eliminated. J. H. Chadwick in [15] analyses in detail the design requirements of the fin servomechanism subsystem including the hydraulic transmission and pump considerations, etc.

E. COMPUTER STUDY OF A FIN STABILIZER SYSTEM

A convenient way to analyze the effectiveness of the fin stabilizer system is by comparing the roll angle ϕ_{off} with the fins stowed assuming that the system is linear.

From Equations V.10, V.11, and V.17 it is apparent that:

$$\frac{\phi}{\phi_{off}} = \frac{1}{1 + Y} . \qquad (V.19)$$

Equation V.19 shows that in order to improve the roll reduction, the loop ratio Y, must be increased. But this loop ratio is severly limited by the resolution of the instrumentation involved. Hence the improvement of the roll reduction can be achieved by adjusting the sensitivity control instrumentation gains. It is pointed out [4], that in general although the acceleration feedback might be used to improve the performance at high frequencies, it probably is not necessary provided that the other parts of the system are well designed. Therefore the acceleration gain k_3 can be set equal to zero and then under that situation equation V.18 becomes:

$$\frac{\phi}{M_W} = \frac{1}{(K_s + \text{Alpha } k_\alpha k_f) + (B_s + \text{Beta } k_\alpha k_f)S + J\ S^2} \qquad (V.20)$$

where Alpha and Beta correspond to the adjustable gains k_1, and k_2.

Since these are two variable parameters, it is convenient to analyze the system using "Parameter Plane" technique, this consists in a graphical method that gives a set of curves as function of the two parameters, the curves are the following:

Constant zeta curves (maps of radial lines in the s-plane) for the range of frequency W_n, specified by the user.

Constant W_n curves (maps of circles centered at the origin of the s-plane) for a pre-programmed set of zeta values.

Constant sigma curves, each one of which is the map of a specific point in the real axis of the s-plane.

Constant zeta-W_n product curves.

The model to be analyzed in this work is a roll damping system designed for a passenger ship whose specifications were determined by Vosper Ltd., Portsmouth, England [16]. The vessel has a pair of fins, each one having an area of about 30 ft^2 and mass-moment of inertia of 177 lb-ft-sec^2. The servo must be capable of simple harmonic motion from hard up to hard down (60 degrees total) in 2 seconds. The moment of inertia (J_s) of the ship about its center of gravity is approximately equal to 56,000 ton-ft-sec^2 ± 20%, the natural damping factor (B_s) of the hull is approximately 2,000 ton-ft-sec, and the effective spring constant (K_s) of the ship is approximately equal to 18,000 ton-ft/rad. Experimental results gave the value of the constants k_α and k_f as 40 rad/volt and 2.540 ton-ft/rad resepctively.

Substituting in Equation V.20 the appropriate values given above, the following equation results:

$$\frac{\phi}{M_W} = \frac{1}{(1.8\text{x}10^4+\text{Alpha}\text{x}2500\text{x}40)+(2\text{x}10^3+\text{Beta}\text{x}2540\text{x}40)S+5.6\text{x}10^4S^2} \tag{V.21}$$

Figure 26 is the parameter plane plot for the model system represented by Equation V.21.

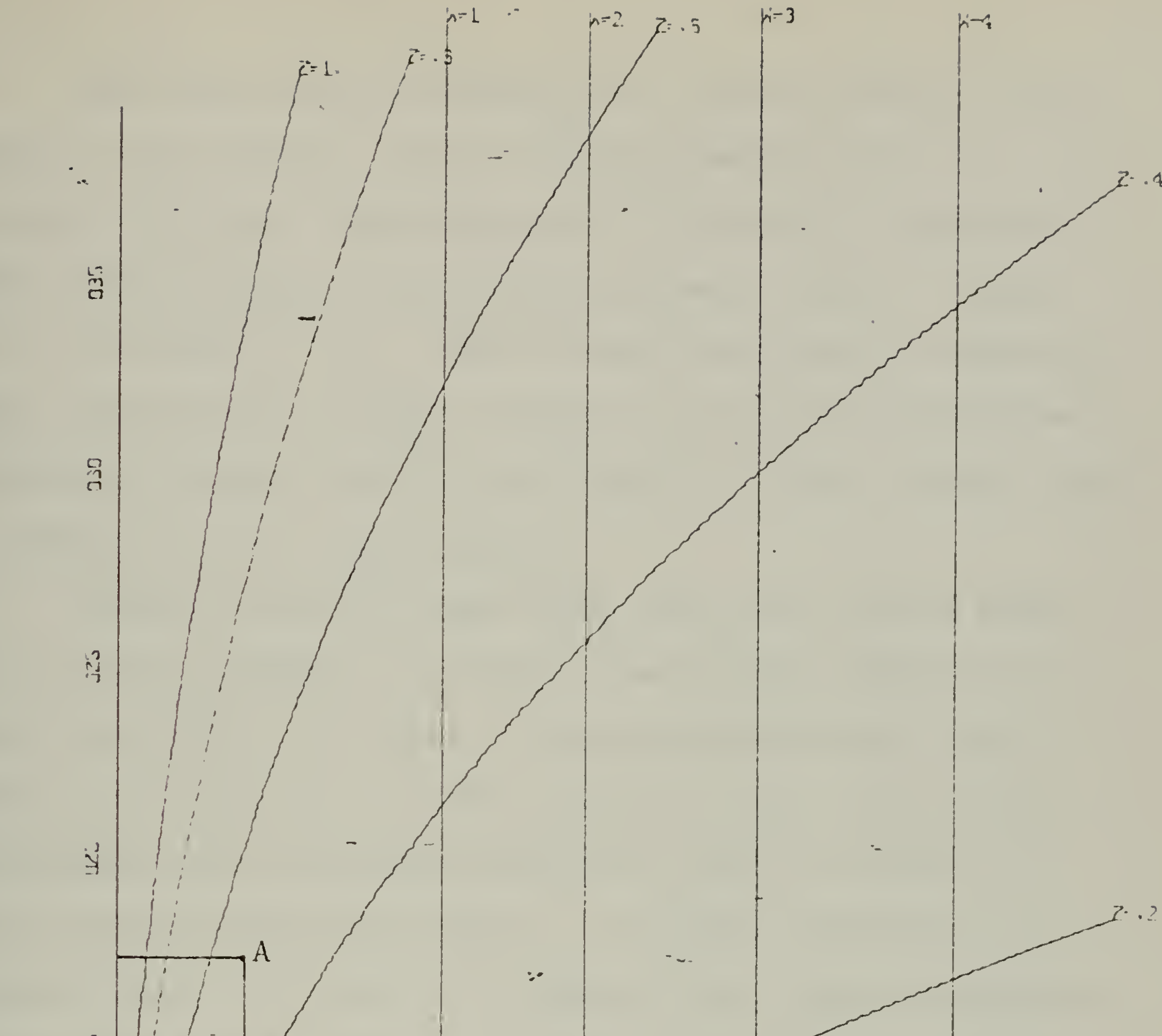

Figure 26. Parameter Plane. Alpha=k_1. Beta=k_2. k_3=0.

With the natural frequency, W_0, of the system calculated as 0.57 rad/sec, and the natural damping factor ξ_0 set equal to 0.5, the operating point "A" shown in Figure 26 was chosen giving the following parameter values: Alpha = 5.0 and Beta = 1.75. These values were used to simulate the system in the digital computer using Continuous System Modeling Program CSMP 360, and Figure 27 shows the unit step response for the example given.

Another study was done using the value of Beta equal 1.75 chosen before, as a fixed value for the velocity sensitivity gain k_2 in order to obtain the parameter plane shown in Figure 28, in this parameter plane, the variable parameters are the angle sensitivity gain k_1 and the acceleration sensitivity gain k_3. In Figure 28 can be observed that for values of k_3 greater than zero the damping factor decreases, which is an undesired situation for the system, and for negative values of k_3 the damping factor increases but the resultant positive feedback could introduce undesired oscillations in the system. Therefore it can be concluded, that as it is discussed in Reference 14 it is not necessary to include in the system the acceleration gain sensitivity k_3 and therefore it will be considered equal to zero for a better performance of the system.

F. CONCLUSIONS

The computer study in this chapter demonstrated the use of parameter plane techniques, as applied to feedback control systems, where all the parameters of the loop had been

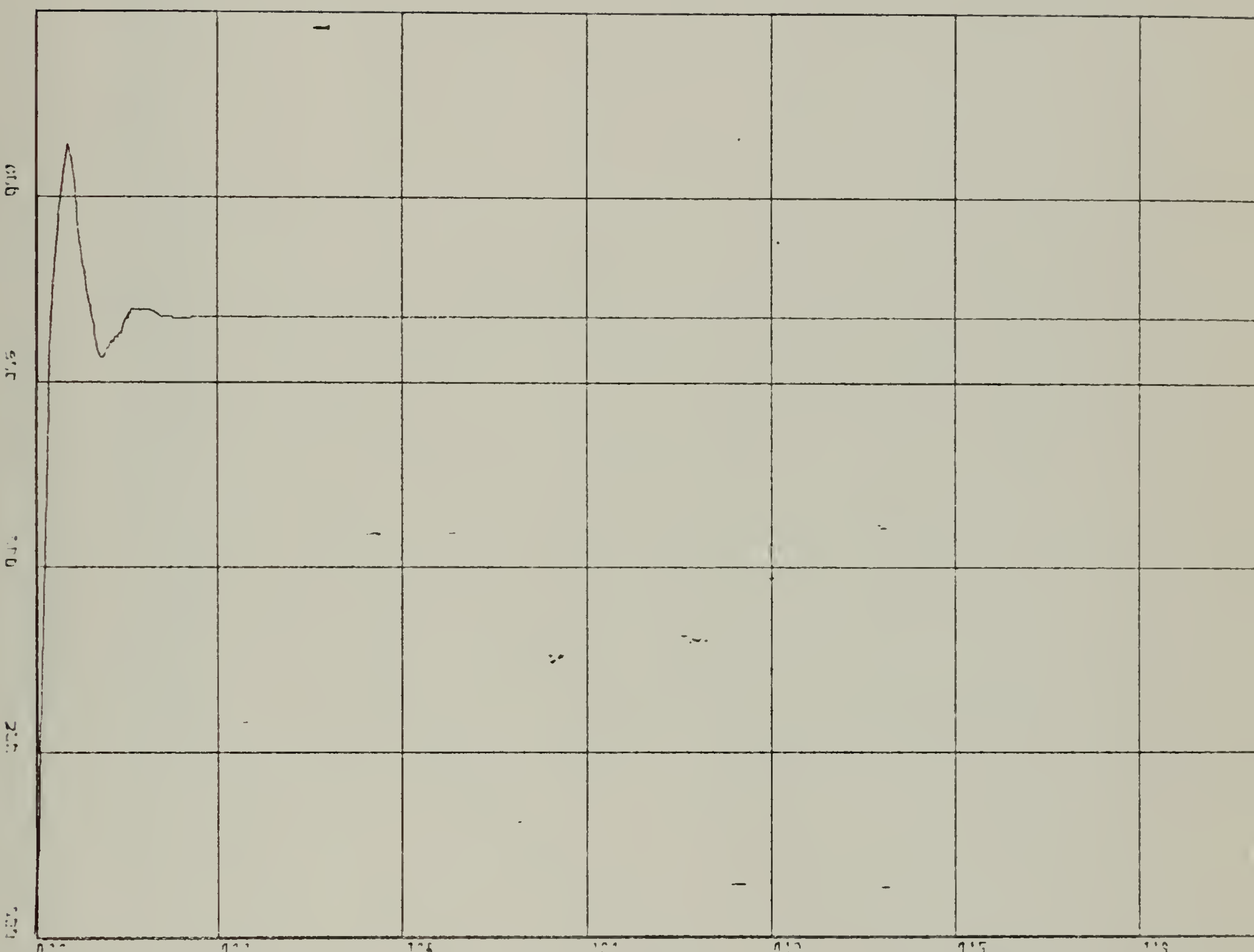

Figure 27. Unit Step Response. $k_1=5$. $k_2=1.75$.

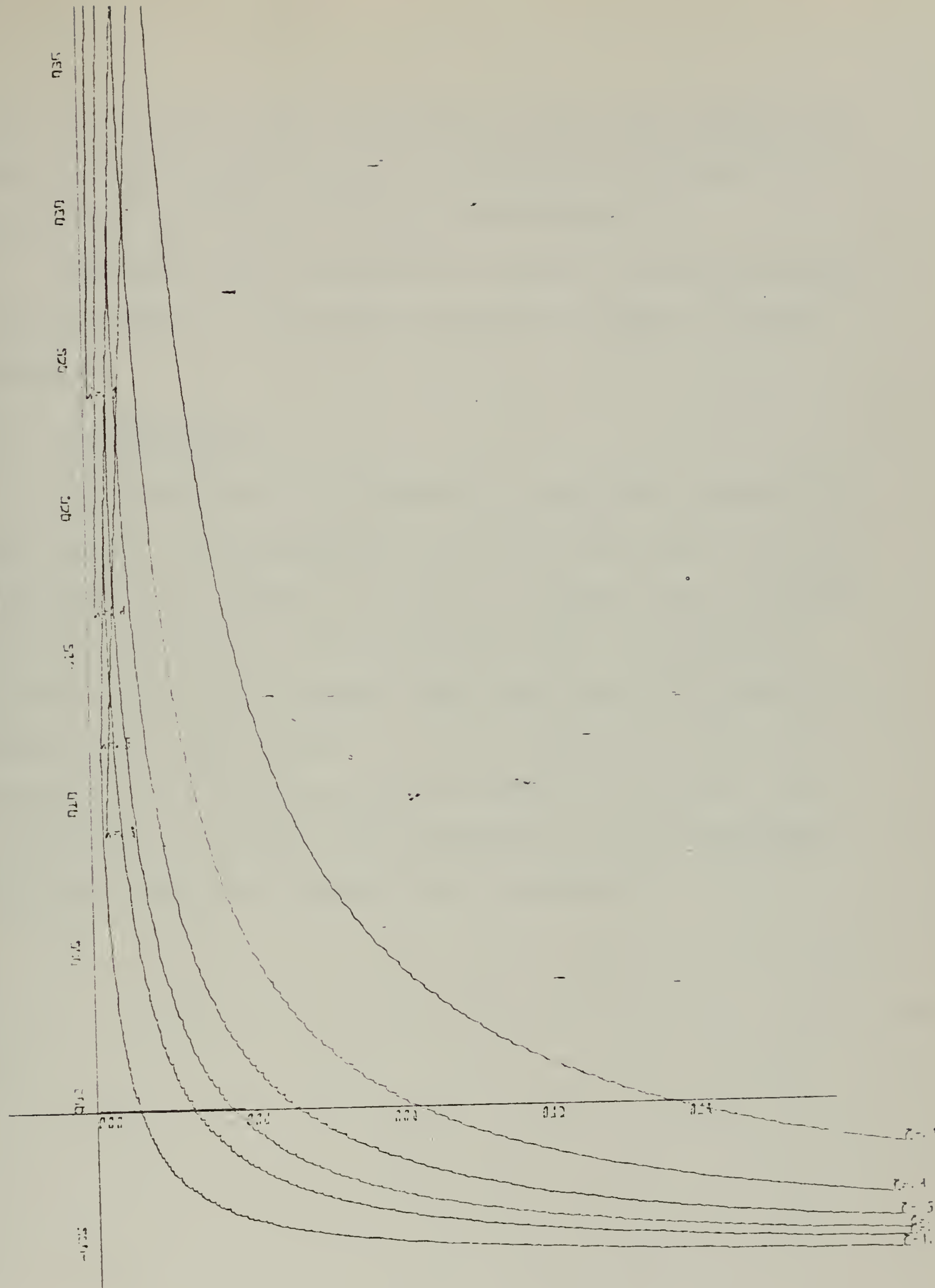

Figure 28. Parameter Plane. Variables k_2 and k_3. k_3=1.75.

selected and the only problem left is the one of determining the appropriate gain values of the control instrumentation to provide an acceptable system performance.

With the aid of the parameter plane, the time expended in field trials to determine the gain settings, is highly reduced.

G. RECOMMENDATION

This study did not consider the constraint imposed by any lag that can be present in the fin servo, and the hypothetical time constant τ_f in the fin dynamics was set equal to zero, therefore it is recommended that the system be studied taking into account these two factors in order to obtain more exact results. On the other hand it was assumed that the system did not saturate at any time, i.e., it behaves as a linear system therefore it is recommended to study the system without this assumption.

VI. A PROPOSED SYSTEM FOR ACTIVATED TANK STABILIZER USING FLUIDIC CONTROL DEVICES AND COMPRESSED AIR

A. FLUIDIC CONTROL DEVICES

The study of the subject of fluidic control systems has shown some advantages over conventional control systems, such as electrical controls or servohydraulic systems. Some of the advantages observed are: a) the absence of moving parts in the system, and b) sea water may be used as fluid of operation either as power fluid or as control fluid because the sea environment does not affect the system thus simplifying the design in comparison with other systems which use different fluids, e.g., oil used in hydraulic control systems [10].

Fluidic control devices operate based on the interaction between moving streams of fluids, either liquid or gases.

These fluidic devices are usually classified as either digital or proportional devices, depending upon the output change which results from a change of the input signal. Digital devices are based on the condition that the power stream normally flows entirely or not at all into an output receiver, i.e., an "on-off" or digital description, which means that the device has distinct output states. Proportional or analog devices have an output which can assume an infinite number of different states in response to very small changes of the input signal. The output changes can be so small that they cannot be measured.

Due to the requirement of passages for the fluid, all the fluidic devices must be three-dimensional, but they are often denoted as "planar" or "axisymmetric." A planar structure is basically a two-dimensional arrangement which has been moved along a line perpendicular to the surface to provide depth, the third dimension. All the planes passing through the element, perpendicular to this line, will have an identical two-dimensional shape described on their surface. The majority of fluidic devices are of the planar type. The axisymmetric type indicates that the shape of the device is symmetric with respect to its centerline or axis of rotation. All planes which include the centerline passing through the device, will have an identical form on their surface.

Three basic types of fluidic devices are known at the present, they are:

1. Boundary layer.
2. Vortex.
3. Stream interaction devices.

Stream interaction devices will be described in this work and more specifically the type known as a "Proportional Amplifier."

Figure 29 represents a stream interaction proportional amplifier. Port P is the input for the power fluid, the ports A' and B' are receiving apertures where the amplified control signal is measured, ports A and B are the inputs for the control signal and the two parts designated with 0,

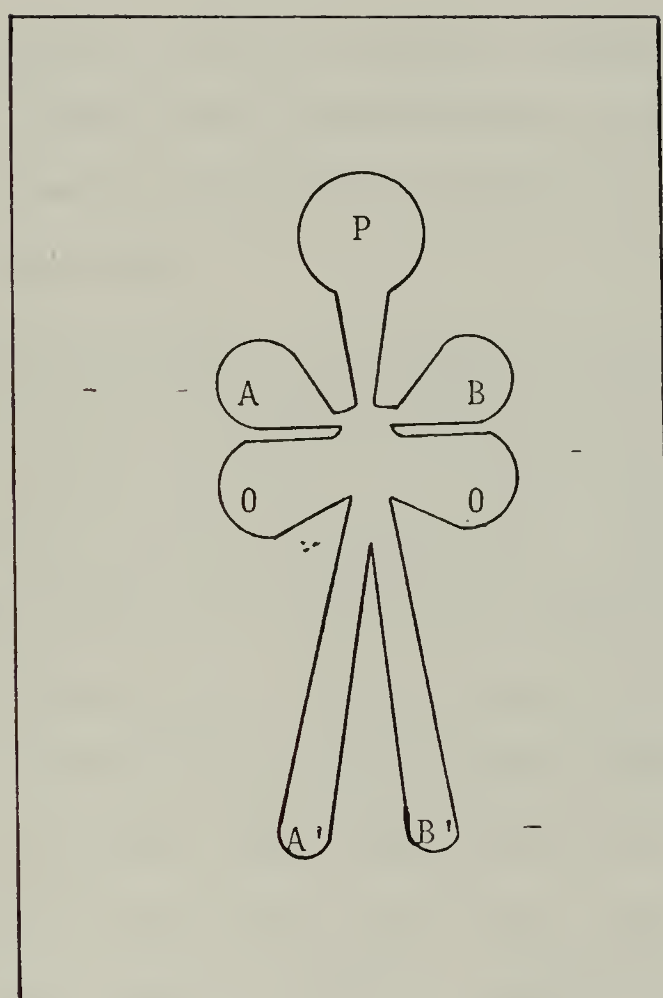

Figure 29. Proportional Amplifier.

are venting ports whose function is to prevent any boundary layer effect that may occur.

If the input control signal differential across ports A and B is equalized then the control device is in equilibrium and the power stream through the nozzle p, divides evenly between the output ports A' and B', or any combination of the two.

This combination of flow produces output characteristics at A' and B' which are proportional to the input characteristics at A and B, from this property arises the name of proportional amplifier.

B. DESCRIPTION AND OPERATION OF THE SYSTEM

The application of reservoirs containing movable fluids whose displacement from side to side in a vessel produce a stabilizing moment that tends to reduce the rolling action, have brought advantages and shortcomings; in passive tank stabilizers for example there exist some limitations in their use, one limitation that under some particular conditions they help to roll the vessel more than the vessel should roll without tanks. But the reduction of roll with the use of tank stabilizers is still a solution of the problem, therefore efforts have been directed toward conceiving a system which will reduce the adverse effects with a minimum consumption of power and with a minimum of complexity.

Activated tank systems have been described in this thesis and in reference [11] is described a system consisting

of two tanks which are interconnected by a fluid channel and contain a reversible propeller whose direction and rate of rotation governs the displacement of fluid from tank to tank.

Obviously an activated system like the one described above requires a very fast response to produce effective stabilization, but it is difficult to meet this requirement because this implies that a column of stationary fluid must be moved as rapidly as possible, and this means a disbursement of relatively high energy.

The operation involved in the described fluidic control device seems to be very suitable for the development of a system which would perform the control of the displacement of fluid in an active tank stabilizer arrangement, this proposed system is shown in Figure 30 with the following description:

The system is composed of two tanks and the interconnected canal, as the system already described, that is, a "U" tube configuration.

Each tank is provided with two valves, one is used for the connection of air pressure into the tank when the water ballast must be transferred to the other tank, and the other is used to extract air from the tank when the water ballast is transferred from the other tank. Both valves are also connected to the appropriate air subsystem arrangement and will be governed by a few stages of fluidic proportional amplifiers which are provided with the required streams

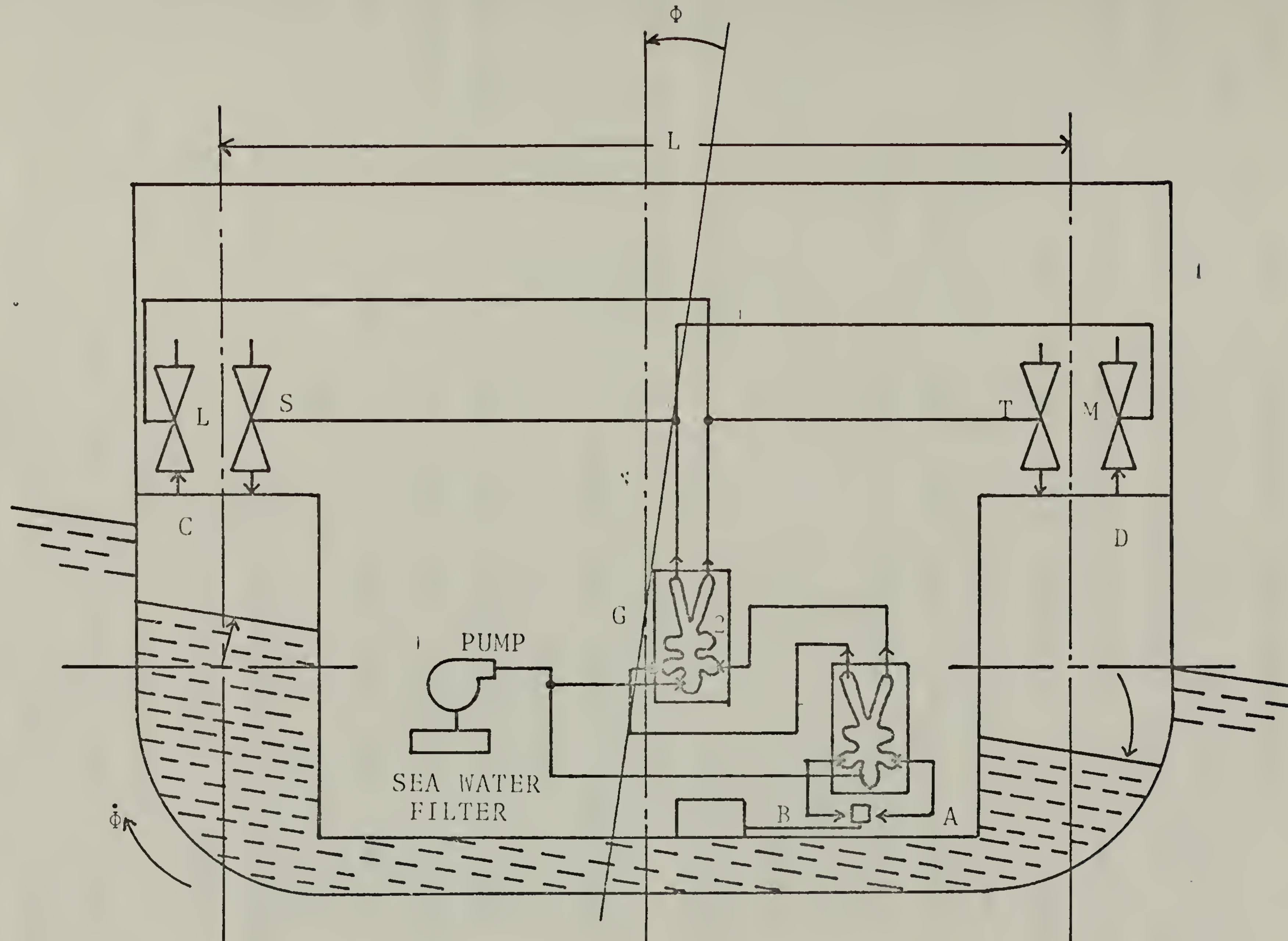

Figure 30. Schematic Diagram of the Proposed System.

of fluids for their respective interaction, the power fluid is provided by a water pump as shown in Figure 30, whose output is directed to the input of the amplifiers.

The system must be provided with a roll sensing device capable of sending a signal to the control loop of the first amplifier in order to produce the required control action.

Refer to Figure 30 to see the operation of the system. At the given instant of time, the ship is considered to have a roll angle ϕ to port side and a clockwise motion angular velocity $\dot{\phi}$ from port to starboard. The angular velocity $\dot{\phi}$ at the given instant of time is increasing in magnitude and should reach its maximum value when the roll angle ϕ is approximately equal to zero, i.e., approximately when the ship passes through its upright position.

The mass of fluid is transferred from the right tank D to the left tank C across the connecting canal, so that the weight of the displaced fluid produces a stabilizing moment which is opposite in sign with respect to the angular velocity $\dot{\phi}$. When the inclination took place, the roll sensing device sent a signal to the control loop of amplifier 1 (see Figure 30) thus producing a proportional change at 0, resulting in a pressure uild-up in line A. The amplified jet will be then deflected to port b' and it is further amplified in the next stages, this output is going to actuate simultaneously on valve T, so that air is pumped into the right tank D, and on valve L to open the air output from the left tank C. This simultaneous operation

of valves T and L result in the reduction of the air pressure in tank C, while in tank D the pressure increases, thus a pressure difference is created at the top of the tanks which finally provides the required force. With the above considerations, the same situation occurs with valves S and M when the inclination and action of the controls are opposite to the previous one described.

C. THE CONTROL ACTION

In general the object of any roll stabilizer is to produce a stabilizing moment, M_s. This stabilizing moment is given in [12], according to the following equation:

$$M_s = - K_s \frac{d\phi}{dt} \tag{VI.1}$$

where K_s is the equivalent coëfficient characterizing the action of the stabilizer.

As was discussed in the previous section, the concentration of the water ballast must be a maximum in the tank which is instantaneously rising in space at the fastest rate which generally occurs when the ship passes through the vertical position (ϕ=0°). When the ship reaches the extreme position (ϕ_{max}) and $\dot{\phi}$=0 the volume of water in both tanks must be the same, see Figure 8.

Physically the above discussion can be seen from the fact that in all cases the excess of the fluid is on the rising side of the vessel and the energy of the wave slope instead of accelerating the ship and building up larger

angles of roll is continuously expended in lifting an extra weight.

In Figure 8 it is observed that in the action of the tanks to reduce the roll, there is a phase difference of 90° between the ship and the water in the tanks, so that the water in the horizontal leg always runs downhill and thereby creates a damping moment resisting the roll of the ship. This is the case in a passive tank system in which no external power is required and the flow of the fluid is, to a great extent, due to the potential head caused by the difference in heights of the tanks in space due to residual rolling. The active tank system on the other hand increases the rate of transfer of the fluid of the equivalent passive system but does not change significantly the phase of the fluid.

In conclusion, in all the possible situations of rolling, the control of the fluid must be such as to fulfill the condition given in Equation VI.1, therefore the fluidic control system suggested before must fulfill this fundamental condition.

Referring to Figure 30, the stabilizing moment of the anti-rolling tanks is obtained by a variable amount of water W, and a fixed lever arm L, the horizontal distance between the line of centers of the vertical legs of the U-tube. It follows from the above that the stabilizing moment is given by:

$$M_s = WL = K_s \frac{d\phi}{dt} . \qquad \text{(VI.2)}$$

Solving Equation VI.2 for W, gives:

$$W = \frac{K_s}{L} \frac{d\phi}{dt} . \qquad (VI.3)$$

From Equation VI.3 it can be seen that the amount W of the ballast water concentration must be proportional to, and in phase with the instantaneous angular velocity of motion. In this case a primary control of the system must be derived from an instrument responsive to angular velocity such as a constrained gyroscope suitably positioned to respond to rolling.

If Equation VI.3 is differentiated, the result is

$$\frac{dW}{dt} = \frac{K_s}{L} \frac{d^2\phi}{dt^2} . \qquad (VI.4)$$

From Equation VI.4 is observed that another possible primary control of the rate of transfer dW/dt can be achieved by means of an accelerometer.

This procedure can be extended, and additional methods of control can be determined which will be responsive to still higher time derivatives of residual rolling and yet capable of producing stabilizing action satisfying the condition given in Equation VI.1.

D. ROLL SENSING DEVICE AND CONTROLS

If the roll stabilization on ships is considered as a problem in which any acceleration of roll on the vessel should be opposed by a moment in the opposite direction, then this acceleration or a higher derivative should be

used to control the fluid that is flowing from one tank to the other.

But if it is considered as a problem of damping an existing roll, then the true damping will be effected by opposing the velocity of roll. It must be observed that in order to integrate the gravitational force due to the fluid accumulation in one or the other tank a certain amount of time is required, therefore the most convenient way to obtain acceleration, the natural control function, is by differentiation of the velocity of roll.

From the previous discussion it is apparent that a device capable of sensing the roll rate is necessary, and then its output may be differentiated to obtain acceleration.

The most practical and economical device for this purpose appears to be a single-degree-of-freedom gyro, provided with a spring restraint and viscous damping to avoid oscillation.

The use of the gyroscopic instrumentation to sense the rolling angle is going to provide a signal given by the following expression:

$$R_t = g_1\phi + g_2\dot{\phi} + g_3\ddot{\phi} \qquad \text{(VI.5)}$$

where g_1, g_2, and g_3 are respectively, the sensitivity to roll angle, roll rate, and roll acceleration. In general at the frequencies of interest the lags or resonances of the sensing instrumentation can be ignored. The resulting transfer function H will be:

$$H = \frac{R_t}{\Phi} = g_1 + g_2 S + g_3 S. \tag{VI.6}$$

Once the signal R_t is obtained, it is amplified through an adequate number of fluidic amplifier stages and provides the necessary energy to actuate the proper valve arrangement in the system.

E. MATHEMATICAL MODEL FOR THE SYSTEM

The combined motion of a ship with a tank stabilizer is represented by the following two linear differential equations:

$$J_s\ddot{\phi} + B_s\dot{\phi} + K_s\phi - P_{st}\theta - Q_{st}\ddot{\theta} = K_s \text{ a sin wt} \tag{VI.7}$$

$$J_t\ddot{\theta} + B_t\dot{\theta} + K_t\theta - P_{st}\ddot{\phi} - Q_{st}\phi = 0. \tag{VI.8}$$

Equation VI.7 represents the ship dynamics with the coupling coefficients between ship and tank represented by P_{st} and Q_{st}, and Equation VI.8 represents the stabilizer dynamics considering also the two coupling coefficients P_{st} and Q_{st}. Equations VI.7 and VI.8 determine a system totally passive in which the stabilization of roll is accomplished by the action of weight of the displaced water from tank to tank, therefore in order to have a representation of the suggested activated tank stabilizer, it is necessary to introduce in Equation VI.8 another factor which represents the effect of the control instrumentation, certain delay due to the opening and closing of the air valves in addition to the gain K_A of the fluidic amplifiers, assuming that the amplifiers are ideal, i.e., having no delays.

The roll-sensing instrumentation was defined before and will be represented by the Equation VI.5. The amplifier gain K_A and the valve arrangement will be combined and can be given by the following expression:

$$G_v = \frac{F_v}{R_t} = \frac{K_A}{1 + \tau_v S} \qquad \text{(VI.9)}$$

where F_v is the required signal to actuate the air system in order to generate the stabilizing moment, and τ_v is the time delay assumed for opening and closing the valves in the system.

From the above considerations, Equation VI.8 must be modified introducing the term f_v in the right side of the equation and the resultant equation is given by:

$$J_t\ddot{\theta} + B_t\dot{\theta} + K_t\theta - P_{st}\ddot{\phi} - Q_{st}\phi = f_v. \qquad \text{(VI.10)}$$

F. THE SIMULATION

The simulation attempts for this system were conducted using a model ship under conditions which were calculated to show its effectiveness as compared with a conventional passive system.

Equations VI.7 and VI.10 representing the proposed system were used as the basis for the simulation, and for purposes of comparison, the conventional passive system was simulated setting f_v equal to zero.

The system was simulated in the digital computer using the Continuous System Modeling Program CSMP 360 and the Equations VI.7 and VI.10 were written in a more convenient

form in order to realize the simulation. Equations VI.11 and VI.12 are the equations used to do the simulation:

$$\ddot{\phi} = -\frac{B_s}{J_s}\dot{\phi} - \frac{K_s}{J_s}\phi + \frac{P_{st}}{J_s}\ddot{\theta} + \frac{Q_{st}}{J_s}\theta + IN \qquad (VI.11)$$

$$\ddot{\theta} = -\frac{B_t}{J_t}\dot{\theta} - \frac{K_t}{J_t}\theta + \frac{P_{st}}{J_t}\ddot{\phi} + \frac{Q_{st}}{J_t}\phi + f_v \,. \qquad (VI.12)$$

When the simulation was first attempted, an implicit loop prevented solution. In order to avoid this problem, a real pole was introduced whose effect in the system can be neglected.

The simulation was realized according to the block diagram shown in Figure 31 which is a representation in block form of the different terms considered in Equations VI.11 and VI.12 and the real pole mentioned above. The different blocks are given in a convenient way to use the Continuous System Modeling Program mentioned before.

G. THE MODEL SHIP AND TANKS

The model ship used to simulate the system is one given in Reference 3 for purposes of solution of numerical examples and the parameters and values determined for its use were the following:

Displacement weight W = 936 Tons

Metacentric height GM = 0.73 m.

Ship Moment of inertia J_s = 1290 Ton-m-sec

Natural Damping Factor of Ship B_s = 60 Ton-m-sec

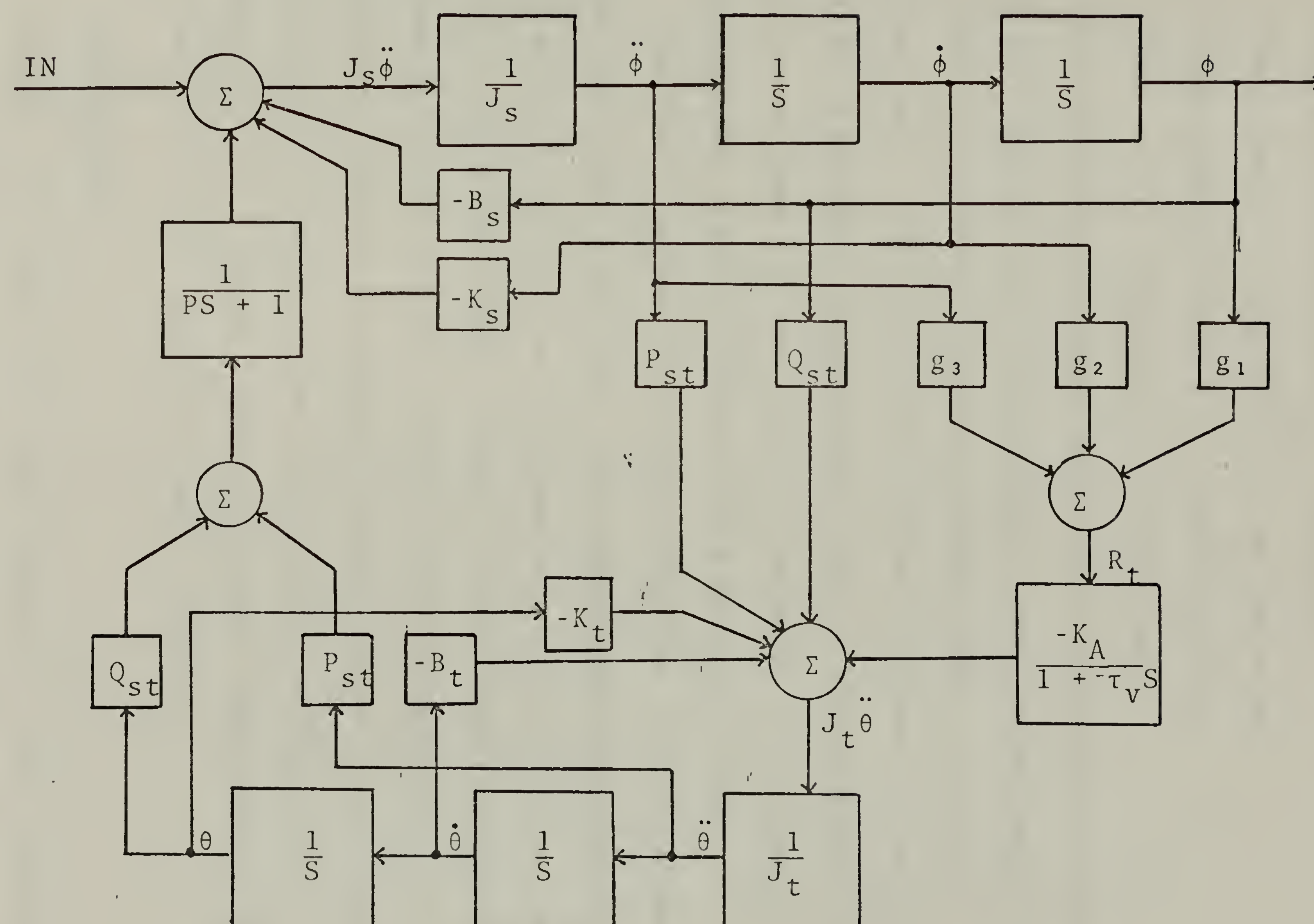

Figure 31. Block Diagram. Activated Tank Stabilizer Simulation.

Ship spring constant = K_s = 684 Ton-m.

Coupling coefficient P_{st} = 24.2 Ton-sec

Coupling coefficient Q_{st} = 76 Ton.

Tank Moment of inertia J_t = 38 Ton-m-sec

Tank Damping factor B_t = 0.60 Ton-m-sec

Tank spring constant K_t = 370 Ton-m.

The tank coefficients given above were calculated according to the criterion and specifications given in Reference 18, and some of the basic design considerations are fairly self-evident. Utilizing the maximum ship beam for example results in maximum effectiveness, and if the tanks are located at or near the yaw axis of the ship, this reduces some coupling effects between fluid motion and yaw.

The basic parameter in tank selection is the volume of fluid the system is to use. The specifications given in Reference 18 indicate two fundamental points for the tank design. The first point is that a total fluid weight of between 1/2% and 1-1/4% of total ship displacement is sufficient to produce an effective stabilization if the tanks are properly arranged. The second point is that for a steady list of 1 degree, the tank static moment should be at least 12% of the moment to heel 1 degree.

H. THE RESULTS

Figure 32 is the unit step response for the simulation of the passive tank stabilizer system, this was realized setting the right hand term of equation VI.8 equal to zero,

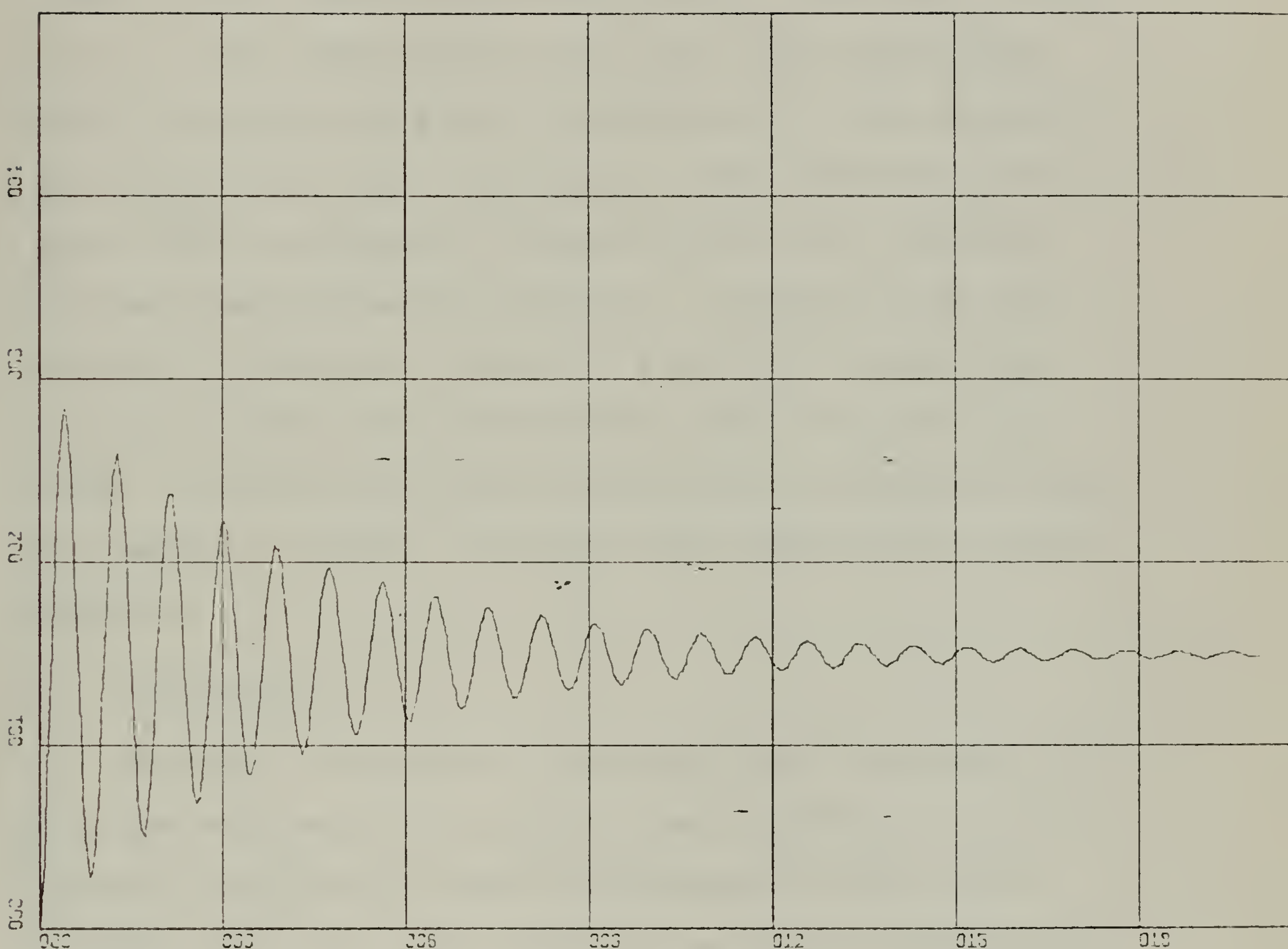

Figure 32. Unit Step Response. Passive Tank Stabilizer.

and it was referred as the point of comparison with the proposed system.

Figure 33 is the unit step response for the simulation of the proposed activated tank stabilizer system, the values of the control gains used were $g_1 = 0.50$, $g_2 = 1.25$ and $g_3 = 2.40$, these values were taken from experimental results obtained and given in Reference 18. The amplifier gain used in the first run was $K_A = 1000$. Observe in the Figure that the damping is greater than in the previous case and the oscillations reduced. In Figure 34 the step response is shown as a result of a run with the same control gain values but increasing the amplifier gain to $K_A = 10000$. It can be observed that with the new amplifier gain the damping is further increased thus improving the system behavior.

I. CONCLUSIONS

The major conclusion to be drawn from this chapter is that the performance of a passive tank stabilizer system could be significantly improved through the use of the proposed activated tank stabilizer, as it was demonstrated by the results obtained in the computer simulation realized, i.e., the transient damping was increased when the activated system was used instead of the passive system.

J. RECOMMENDATIONS

It is recognized that the passive tank stabilizer was designed to reduce the rolling action effectively without

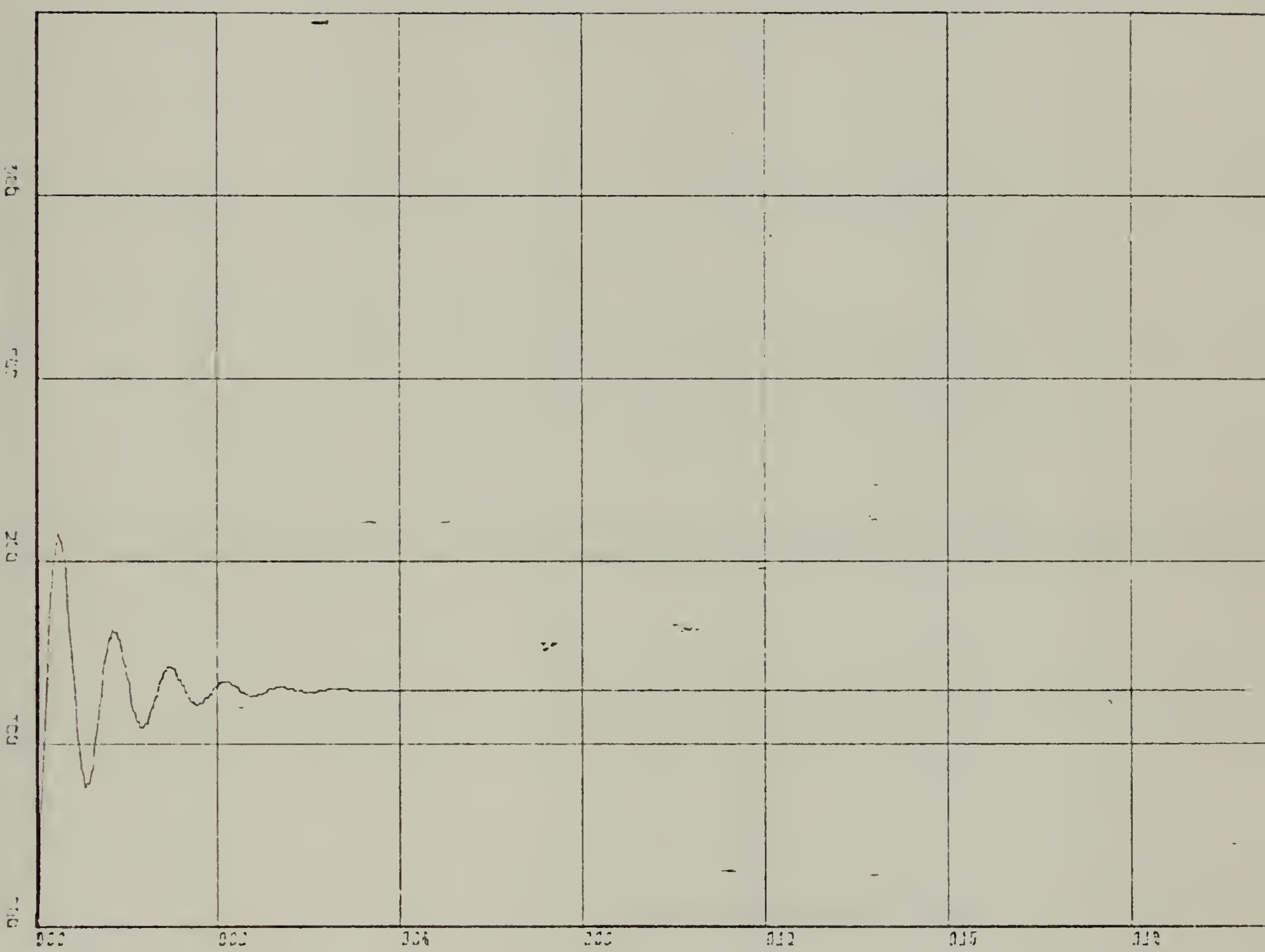

Figure 33. Unit Step Response. Activated Tank Stabilizer.
K = 1000.

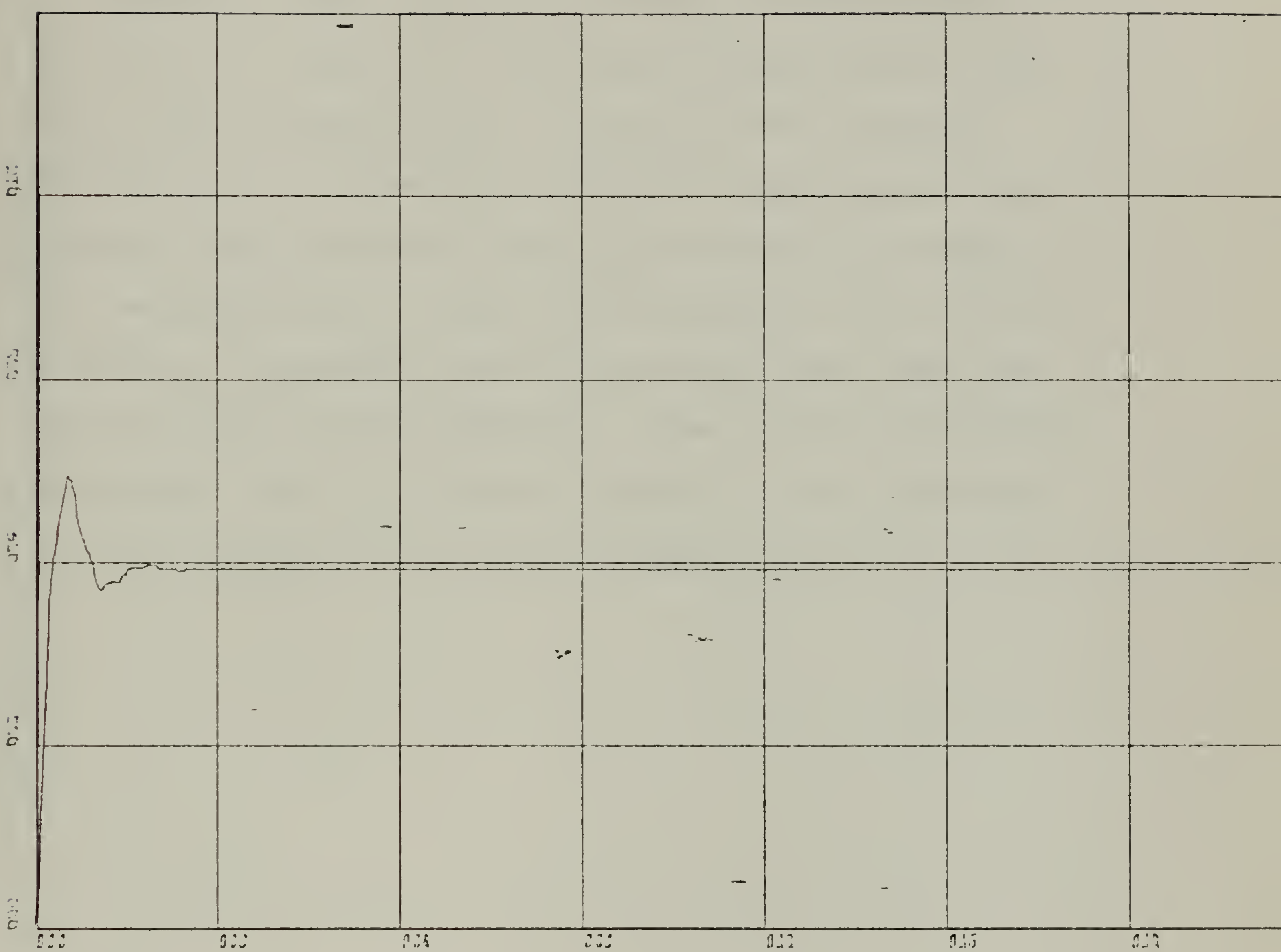

Figure 34. Unit Step Response. Activated Tank Stabilizer.
K = 10000.

expending a significant amount of power. Furthermore, it is realized that any proposal for increasing the roll stabilization of any ship must account for most efficiency in its operation. With this in mind the system previously described was proposed. The study realized does not mention certain aspects of the system and their effects. Therefore it is recommended in future investigations to realize a more complete study that includes for example the characteristics of the valve arrangement and the compressed air subsystem, and the constraint that they can impose in the system's behavior. Some of these characteristics are: the air pressure, capacity of the subsystem. size and weight of the system's hardware, etc.

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13. ABSTRACT

The theory of Roll stabilization of Ships is presented in the context of modern control theory. The most common systems used to reduce the roll are described, and the principal equations are formulated.

A general approach for the analysis of roll stabilizers is developed and it is applied to an activated fin stabilizer system. For this approach parameter plane techniques were applied, and the system was simulated in the Digital Computer by means of the Continuous System Modeling Program CSMP-IBM/360.

Finally a system is proposed which is intended to improve the performance of passive tank stabilizers introducing fluidic devices in the feedback loop in addition to a supply of air compressed to actuate on the water ballast. The system was simulated using the same program CSMP-IBM/360, and the results compared with those obtained in the simulation of a simple passive tank stabilizer, showing a significant increasing in the damping.

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